POWER TRANSMISSION DESIGN

JANUARY 1959

OEM··MAINTENANCE

REPETITIVE SHOCK...HOW TO COPE WITH IT

ADJUSTABLE-PITCH SHEAVES DIVIDE TORQUE RATIO

POWER DRIVES IN TIGHT SPACES

HOW TO CALCULATE UNBALANCED THRUST LOADS

DOUBLE HELICAL GEARS REDUCE NOISE

HOW TO CALCULATE MOTOR SIZE AND SPEED

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News and ideas for designers and plant engineers who use power drive equipment

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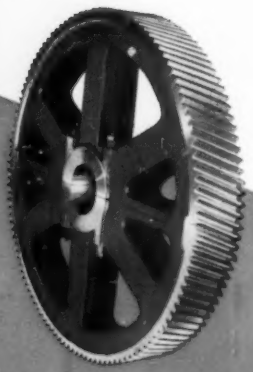

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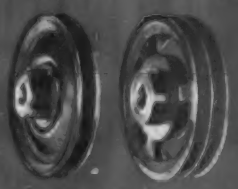

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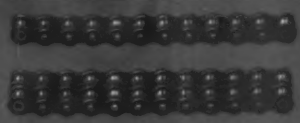

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Goodyear Positive Drive Belts are manufactured under license and according to U.S. Patents 2,507,852 and 2,397,312.

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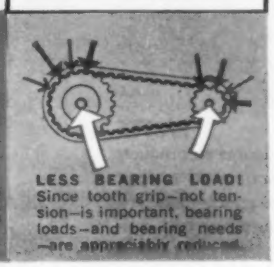

LESS BEARING LOAD! Since tooth grip—not tension—is important, bearing loads—and bearing needs—are appreciably reduced.

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COVER
shows the wide scope of our magazine as it covers its specific field: Power Transmission Design.

POWER TRANSMISSION DESIGN

READERS' GUIDE

January 1959, *volume 1 number 1*

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Driving torque is divided evenly by sheaves with adjustable-pitch characteristics. Hydraulic servos help in accomplishing this.

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Synchronization of speed was impossible until the designer hit on the idea of using a secondary belt and a common power-source.

40 **VARIABLE-VARIABLE DRIVE PROVIDES SECONDARY SYNCHRONIZATION**
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42 **UNITIZED DRIVE ELIMINATES DOWNTIME**
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46 **CHAIN-DRIVEN CONVEYORS ELIMINATE FIELD LABOR**
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47 **HYDRAULIC MOTOR KEEPS FABRIC TENSION CONSTANT**
How may a hydraulic motor stand guard, keeping tension constant under altering conditions? This idea gives the facts, figures, and technical details.

48 **ENCLOSED GEARING STOPS ABRASIVE WEAR**
What can a production supervisor do when the air in his plant contains abrasive grit that chews at gears and pinions? The engineer came up with an idea-answer.

49 **STAINLESS BEARING PERMITS ROTATION AT 1400°F**
High-acid, high-heat conditions call for new developments in bearings for smooth action. Engineers came up with this idea.

50 **HYDRAULIC MOTOR GIVES INFINITELY VARIABLE SPEED CONTROL**
When it is necessary to give a machine-operator complete control over the speed of his machine, it's a good idea to use a hydraulic component.

51 **BELT DRIVE CUTS UPKEEP 95%**
What is the answer when maintenance costs mount? This idea, using a belt drive in place of shafts and right-angle gears brought costs down.

FACTS
and figures on products and literature of interest to designers, engineers, and other power transmission design people can be had by circling the right numbers on the reader service cards found opposite page 84.

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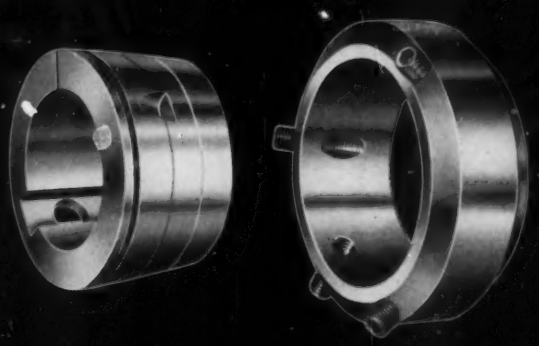

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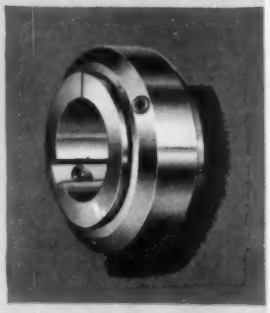

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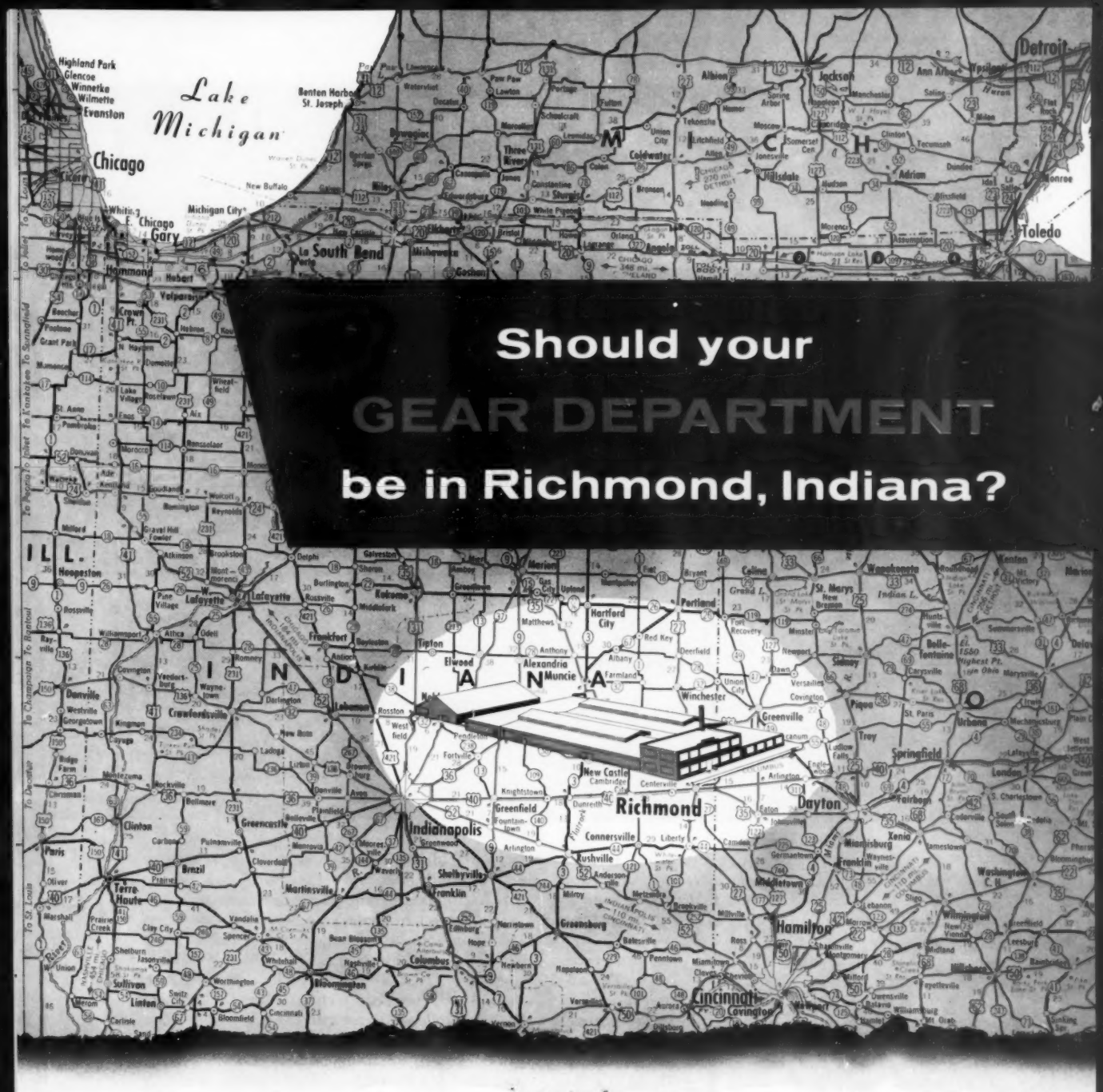

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NEW HIGH CAPACITY!

NEW SMALL SIZE!

MANY NEW SAVINGS!

Dyna-V is entirely new. In every detail, Dyna-V has been engineered to take advantage of today's great improvements in belt materials and metal alloys—resulting in V-belt drives that are sensationally compact.

Dyna-V is capable of handling up to three times as much horsepower in a given space. Dyna-V opens vast new possibilities for better, more economical machine design. And, in most instances *Dyna-V costs less!*

An entirely new high in strength and horsepower capability is packed into Dyna-V Belts. Cross section dimensions are substantially reduced—top width is much narrower. Dyna-V Sheaves are designed for these belts. Narrower grooves reduce face width—and weight. Costs are lowered. Smaller diameter sheaves and shorter center distances multiply savings.

Increased sheave strength for increased capacity is obtained through changes in design that utilize the full benefit of modern improvements in alloy metals. Dodge is noted for the superiority of its semisteel castings, to which now have been added ductile iron castings produced with the most modern electric furnace facilities.

Dyna-V Sheaves are equipped with Taper-Lock Bushings. The superiority of this mounting, patented by Dodge, is widely recognized. Taper-Lock's holding power is terriffic, yet it is "easy on, easy off." And Taper-Lock Bushings provide the additional advantage of interchangeability.

The benefits of Dyna-V go beyond compactness and initial low cost. Smaller sheaves reduce shaft overhang—increase bearing life. The dimensional stability of Dyna-V Belts (which are heat resistant, oil resistant, static conducting) solves the problem of belt matching. Every belt carries its full share of the load. The crowned top of Dyna-V Belts is a new concept in belt design. Combined with proved concave sidewall construction, it insures perfectly even load distribution between belt and sheave.

A complete bulletin on Dyna-V, including drive selection tables, will soon be ready. If you want one of the first copies, write us at once. Or see your local Dodge Distributor.

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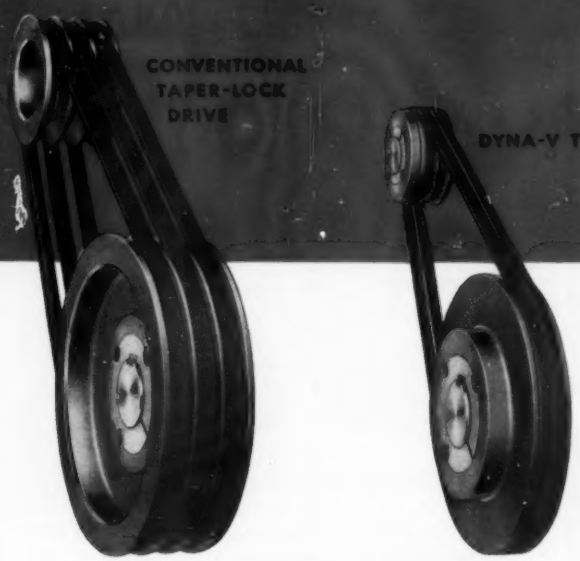

Dyna-V Drives for capacities up to 50 horsepower are now available from stock, using the new belt cross section, "3V," and Dyna-V Taper-Lock Sheaves to match.

Dyna-V Drives for capacities up to 200 horsepower, using "5V" belts and sheaves will be available from stock in March. Larger sizes up to 1500 hp, with "8V" belts and sheaves, can be furnished on order.

COMPARISON, ILLUSTRATED ABOVE — 3 HP, 1750 RPM, 2.41 TO 1 RATIO

	BELTS		SHEAVES			CENTER DIS-TANCE	HP	WEIGHT (LBS.)	COST	
	Number	Size	O.D. Driver	O.D. Driven	Width				Per hp	Per Drive
CONVENTIONAL TAPER-LOCK DRIVE	3	A42S	4.15"	8.95"	2½"	12.3"	4.62	24.7	$5.91	$27.38
DYNA-V TAPER-LOCK DRIVE	2	3V400	3.35"	8.00"	1 3/32"	10.8"	5.0	18.0	$4.50	$22.50
% SAVING *Savings vary with different sized drives*			19%	10%	56%	12%	+	27%	23.8%	17.8%

CALL THE TRANSMISSIONEER—your local Dodge Distributor. Factory trained by Dodge, he can give you valuable help on new, cost-saving methods. Look in the *white* pages of your telephone directory for "Dodge Transmissioneer."

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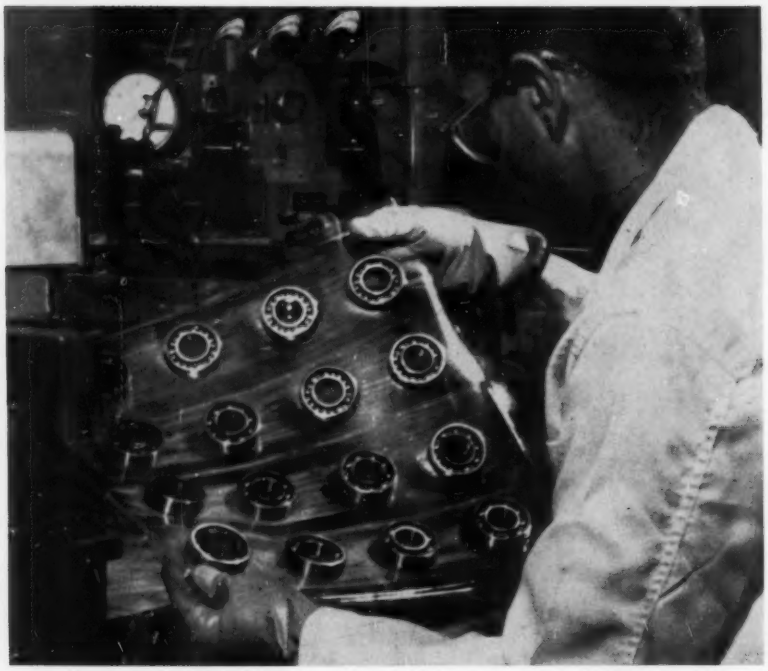
NEWS

from the power transmission field



SIXTEEN BEARINGS have been drape-formed, using the bearings themselves as molds.

Transparent packaging for bearings

WASHINGTON, D. C.—The Department of Navy, Bureau of Ships has unveiled a new wrapping technique for anti-friction bearings.

The materials used are cellulose acetate and cellulose acetate butyrate, because of their extreme compatibility with preservative oils and good storage qualities. Tests aboard two ships over two and half years have shown no bad effects. A preservative oil in accordance with Mil-L-7870 is contained inside the wrap with the bearing.

It was stated that the need for transparent wraps came about as a result of excessive handling for either inspection or identification. Transparent packaging permits identifying the bearing without unwrapping.

The Bureau of ships is preparing Mil-T-2184 to outline this specification. It has not yet been evolved on current procurement contracts. The identifying label is relatively inexpensive and easily handled during high production operations. This contrasts the initial high cost of equipment.

During wrapping, the plastic material is draped over the bearing to afford a skin-tight pack. After the first draw, the material is trimmed so that only half a cup remains, the assembly inverted, preservative added and a second drape applied. This, in effect affords two-cup protection. Sealing is accomplished by dipping the ends in a solution of cellulose acetate and quick-drying solvent. The bearing can be placed in standard cartons.

The label is made by typing the identifying information on albene tracing paper and photographing it. It is pasted on like a postage stamp with an acetone solution. The label goes between the two wrapping layers.

Larger bearings are draped through the inner diameter so that the pack resembles a doughnut. Thas saves on preservative.

The cost of the transparent pack should compete favorably with the cost of other, current packaging methods approved under military specification, MIL-P-197. Estimates of the cost of this pack are about 30 to 40 cents for each bearing.

Broken down, this reveals 20 to 25 cents for the plastic and 10 to 15 cents for the protective carton. When the multiple pack is used with smaller bearings, the cost is close to 15 cents per bearing . . . about 8 cents for plastic, 5 cents for the carton.

AGMA meets with NLGI at fall meeting

CHICAGO, ILL.—The American Gear Manufacturers Association held its semi-annual meeting here during the week of Oct. 27 in joint conference with the National Lubricating Grease Institute. Committee meetings and presentation of technical papers highlighted the work scene.

A very interesting paper was presented to AGMA by G. W. Michalec, General Precision Laboratories, Inc. An abstract of his paper appears in this issue, starting on page 76.

At a general luncheon meeting, John A. Bradner, President, The Lees-Bradner Co., presented Search for Silence. This paper covered the history of gearing

continued on page 83

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in the development of Roller Chain

FIRST TO PRE-LOAD PRODUCTION ROLLER CHAIN

Pre-loading seats pins, bushings and linkplates; takes out initial stretch.

FIRST TO STUDY FATIGUE

First to make design changes in roller chain parts for improved fatigue resistance.

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Pitch holes in DIAMOND chain sideplates are ball drifted for greater fatigue resistance—(DIAMOND patent).

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OIL RESERVOIR

Seam relief bushings in roller chain reduce run-in time and provide better lubrication—(DIAMOND original patent).

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Bushing seams on DIAMOND roller chain are always placed *away* from load-bearing area for improved resistance to wear and fatigue.

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DIAMOND Roller Chains are the product of more than sixty-eight years of specialization in the manufacture of high endurance roller chain. Some of the quality features developed and pioneered by DIAMOND are considered "extra" or "premium" features in other chains. These are standard in DIAMOND Roller Chains . . . and have been for years. DIAMOND will continue to improve the capabilities of its product, assuring you *always* of a thoroughly dependable chain with superior operating characteristics.

Look for the ◇ on the Links

DIAMOND Distributors have all types and sizes of Roller Chain *in stock*, ready for instant delivery. Look under "Chains, Roller" in the Yellow Pages of your Telephone Directory . . . or write to factory for catalog and name of your nearest distributor.

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Curtis joints benefit by a rigid insistence on uncompromising inspection and quality control at every stage of manufacture, insuring minimum backlash.

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NEWS *continued*

T. B. Wood's Opens New Warehouse

CHICAGO, ILL. — Increasing business in the Chicago trade area, was the reason for T. B. Wood's Sons Co., Chambersburg, Pa., manufacturer of power transmission machinery, is opening a branch warehouse at 1500 South Western Ave., Chicago, as headquarters for Richard J. Ward, district manager. Alfred J. Littlefield will be in charge of the warehouse.

The new Chicago warehouse will serve customers throughout the midwestern and north central states.

Dayton Rubber to Build New Plant

SPRINGFIELD, MO.—Construction of a $1,600,000 rubber factory is announced by Dayton Rubber Co., Dayton, Ohio. The new plant will be used primarily for manufacture of mechanical goods.

New Materials for Nuclear Power Plants

ANN ARBOR, MICH.—American Metal Products Co. has announced the formation of the Engineering Science Div. for research, developing and producing new materials for nuclear power plants.

Under the direction of Dr. H. A. Ohlgren, the division will expend its major efforts on high temperature materials. Its laboratory is located at 1355 No. Main St., in Ann Arbor.

Cullman Opens Manhattan Warehouse

NEW YORK, N. Y.— Cullman Wheel Co. announces opening of a new warehouse at 239 W. 18 St. for stocking roller chain and sprockets, especially off-the-shelf items. "800 different items will be handled for instant delivery", cites Norman Wienke, President. The firm's home office is 1344 Altgeld St., Chicago.

Power Shift Transmission Increases Truck Efficiency

DANVILLE, ILL.—A new two speed, hydraulically-shifted transmission with a constant-mesh design and integral torque converter improves a new lift truck efficiency. Two speeds forward and two reverse are selected by the operator with levers. All components are housed as a unit and mounted to the engine as an integral part of the power train.

The torque converter automatically adjusts power output for load and road conditions within each range. The engine always runs at most efficient speed. No braking, load or drive reactions are carried by the transmission. Manufactured by Hyster Co.

Aeroquip Adds Power Transmission Dept.

WEST LOS ANGELES, CALIF. —Marman Div., Aeroquip Corp., has announced the opening of a new power transmission department. All standard sizes of roller chain, cable chain and double-pitch chain are being stocked. A new roller chain catalog is available by writing the 11214 Exposition Blvd. address.

Shafer Switches To Green On All Products

CHICAGO ILL.—Taking notice of the use of finishes as a sales-aid in marketing, product-finishing people of a leading, self-aligning bearing firm gave their company's products a new look.

Shafer Bearing Division of Chain Belt Company, located at Downers Grove, Illinois, has announced that all Shafer Bearing products, which have been painted gray as standard, are being finished in Chain Belt forest green.

The new color will give the firm's products an outstanding appearance. Customers who have been used to the former, standard gray color, will be especially likely to notice the new finish on the outside of these self-aligning bearings which are used in many industries.

There will be no change in the product design, packaging or pricing.

Shafer Bearing is a leading manufacturer of self-aligning roller bearings for industrial and aircraft use.

For more information, circle No. 20 on the Reader Service Card →

NEW! CROWN GEAR UNITS

NOW 17 Models for Right Angle Power Transfer

No other unit has all these engineered features:

- Light, compact, completely enclosed 356 Aluminum alloy housing—designed for maximum strength and heat dissipation.
- Non-magnetic, Stainless Steel Shafts.
- Double Sealed, hardened and ground ball bearings at all Shaft extensions.
- Double Shielded, Hardened and Ground Ball Bearings.
- Hardened Steel, Spiral Gears provide smoother, quieter operation and higher load capacities.
- Choice of Four and Five Mounting Positions to permit neat, compact installation to wide variety of applications.
- Choice of Two and Three Shaft Extensions.
- Ratings from ¼ through 2 H.P.
- 1:1 or 2:1 Ratio optional in all sizes.
- Provision for quick, easy lubrication.
- Constructed to allow easy, inexpensive replacement of any individual component should it be required.

Simplify your right angle power transmission problems by using a Crown Packaged Unit. There's a stock unit among Crown's new line which will fit practically any application. Designed for rugged industrial use, these compact units are completely enclosed, and will provide years of reliable service.

By using a Crown "package" you eliminate much detailed design work, enabling you to devote more effort to your overall design problem. They're the ideal solution to all types of right angle power transfer, and are especially adaptable to confined areas.

SPECIAL CROWN GEAR UNITS MANUFACTURED TO YOUR SPECIFICATIONS.

1. Non-Standard Housing Material.
2. Non-Standard Shaft Material.
3. Non-Standard Gear Material (Formica, Nylon, etc).
4. Special Shaft Extensions.
5. Special Machining on Shaft Extension (Flats, Keyways, etc.)

Check your leading power transmission distributor or write direct

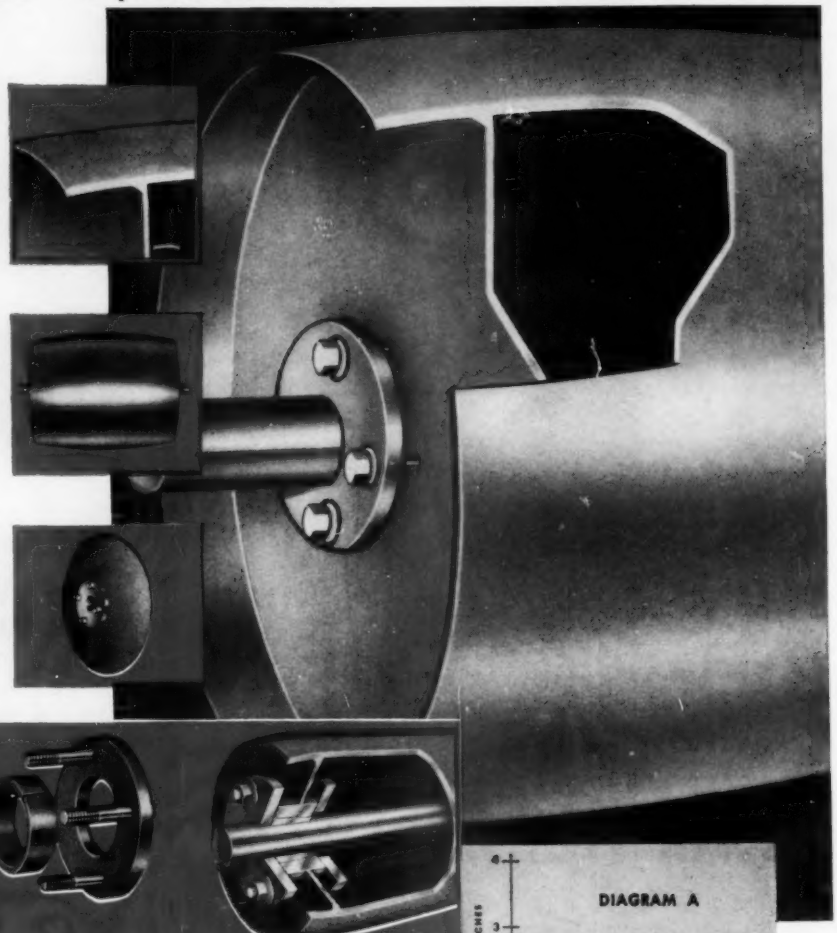
CROWN GEAR
a subsidiary of Harrington & Richardson, Inc.
320 PARK AVENUE, WORCESTER 10, MASS.

SOME DISTRIBUTORSHIPS OPEN

STEPHENS-ADAMSON WELDED ALL-STEEL PULLEYS

CURVE CROWN BY STEPHENS-ADAMSON

FEATURING "SQUEEZELOCK" HUB DESIGN



CURVE CROWN* DESIGN

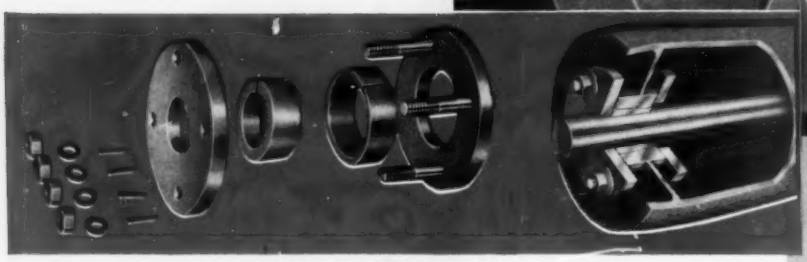
Curve crown on outer ends of rim accurately formed. Revolutionary design eliminates conventional center peak — high point for belt stretch and wear — while increasing belt training effect more than 100%. See diagram (A).

STURDY RIM CONSTRUCTION

Pulley rims are made of one-piece construction, formed absolutely round under hydrostatic pressure. The only seam is machine-welded both inside and out to insure 100% penetration of welds.

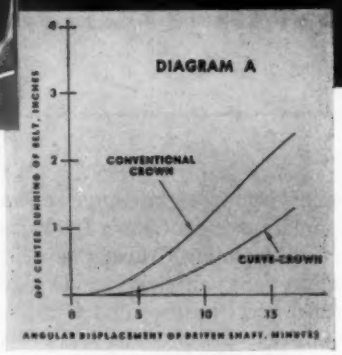
ACCURATE END PLATE ASSEMBLY

End plates are machined on both the O.D. and I.D. to insure concentricity between the bore and the outer rim. They are pressed into position for tight fit-up and submerged arc-welded for maximum efficiency.



"SQUEEZELOCK" HUB

Revolutionary design of "Squeezelock" Hub provides gripping power for full torque transmission without the use of keyways and eliminates distorting loads against pulley end plates. Two split tapered bushings are wedged against shaft and pulley end plates by two independent hub plates which are squeezed together by tightening four large diameter bolts.



*PATENT APPLIED FOR

WRITE TODAY FOR BULLETIN 558

STANDARD PRODUCTS DIVISION

STEPHENS-ADAMSON MFG. CO.

GENERAL OFFICE AND PLANT: 71 RIDGEWAY AVENUE • AURORA, ILLINOIS

PLANTS AT: CLARKSDALE, MISSISSIPPI • LOS ANGLES, CALIFORNIA • BELLEVILLE, ONTARIO

For more information, circle No. 80 on the Reader Service Card

Hoover Ball and Bearing Moves Plant

ANN ARBOR, MICH.—The Hoover Ball and Bearing Co. has completed a move of ite entire Ball and Bearing Div. to a new plant on a 90 acre tract south of the city. Extensive conveyor systems, new method of disposing of steel scrap and a long underground pit mark innovations of the $3,500,000 facility which is air conditioned and modern in all respects. It has 200,000 sq ft of manufacturing and office space.

Ohio Gear Names New Distributors

CLEVELAND, OHIO — Ohio Gear Co. has appointed three new distributors: Bearing Supply and Service Co., Ltd., Winnepeg, Manitoba; Taccone Pneumatic Foundry Equipment Co., Erie, Pa.; and Robert J. Uplinger, Inc., Syracuse, N. Y. All three firms will handle the complete line of gears, speed reducers and other equipment made by the company.

Airborne Names Two Distributors

HILLSIDE, N. J.—Airborne Accessories Corp. announces the appointment of two distributors for commercial airline products: Airwork, Inc., will represent the firm in the East; Southwest Airmotive Co., Inc., will handle the Southwest through its outlets in Dallas and Kansas City. Anglegear right-angle bevel gear units, rotary actuators and other components will be handled.

Acme Chain Appoints Potter and Dugan

BUFFALO, N. Y.—The Acme Chain Corp. announces the appointment of Potter and Dugan, Inc., as their Western New York State representatives. Address is 51 Great Arrow Ave. They offer a full line of roller chain, sprockets and conveyor attachments in addition to a broad engineering service.

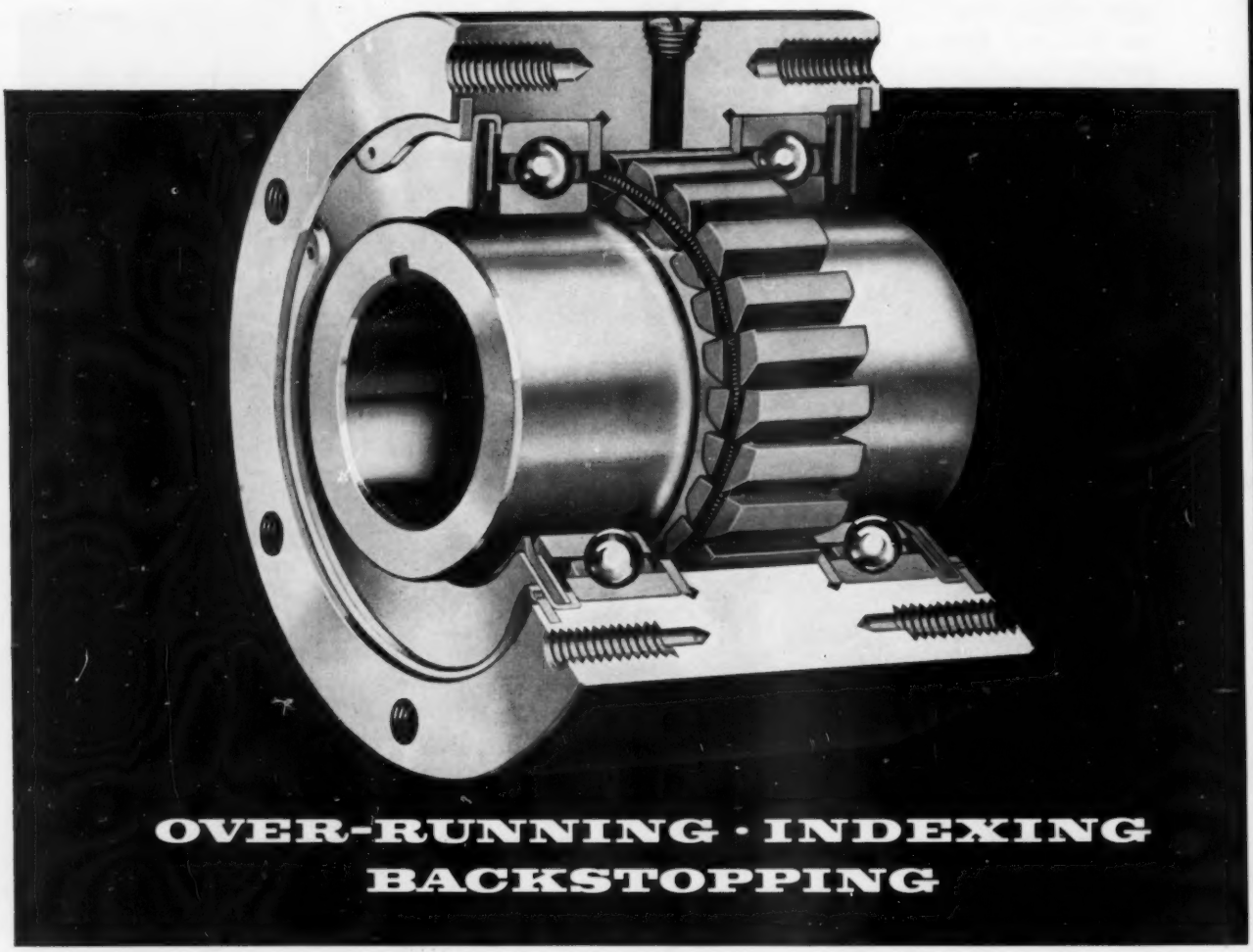
THE BUSHING is the secret! Browning developed a simple, easy-to-assemble *malleable* bushing for interchangeable service in their entire family of sheaves, sprockets, couplings and paper pulleys. The result is many thousands of size and bore combinations right off the shelf, ready to use. Now Browning offers also an outstanding line of bearing units, featuring housings of unbreakable *malleable iron* for utmost strength and compactness. See your distributor for details, or write us for your copy of descriptive Catalog GC101.

Browning Manufacturing Company, Maysville, Kentucky

Browning POWER TRANSMISSION EQUIPMENT

For more information, circle No. 13 on the Reader Service Card

The OVER-RUNNING CLUTCH With

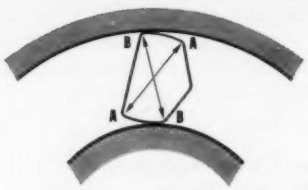


OVER-RUNNING · INDEXING
BACKSTOPPING

STANDARD CLUTCHES FOR ORIGINAL EQUIPMENT NEEDS

Formsprag Clutches are used throughout industry, from business machines to aircraft, in bore sizes from ¼" to 12". Several informative bulletins available, including: "Design Considerations for High-Speed Over-running Clutches," "Ten Design Ideas", "Seven Ways to Reduce Space and Costs", plus the complete Formsprag catalog. Write for any one or all.

HERE'S



The Formsprag clutch consists of a full complement of shaped sprags, or wedges, located between concentric inner and outer races. Power is transmitted from one race to the other by the wedging action of the sprags. Each sprag is so shaped that dimension AA is greater than BB. Rotation of one race in the "driving" direction causes the sprags to wedge, transmitting torque in full from one race to the other.

Greatest Torque
Per Inch – Per Ounce

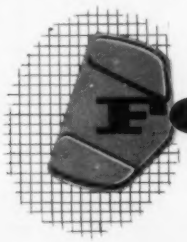
Modern power transmission design demands the utmost torque capacity in an absolute minimum of space. That's why designers in every field specify Formsprag clutches . . . the *modern* way to transmit power on over-running, back-stopping and indexing applications.

Along with highest possible torque capacity for its size and weight, every Formsprag clutch offers such additional benefits as internal simplicity (just four basic parts), no measurable backlash, light weight, compactness, extreme precision and long, trouble-free life. Formsprag's exclusive, patented principle is unlimited in application, yet this full-complement sprag type clutch is extremely simple in design and operation.

There is a size and model Formsprag clutch for every application. Standard models include sleeve bearing, ball bearing, miniature, and large bore backstop types. These are shown in the current Formsprag catalog, complete with specifications, performance data and design suggestions. Ask your Formsprag distributor, or write direct for your copy. For special applications, Formsprag engineers will recommend a modified standard or design a special. Send details of your requirements.

FORMSPRAG COMPANY
23587 Hoover Road, Dept. 100
Warren (Detroit), Michigan

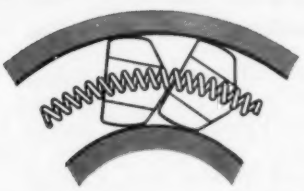
In Canada: Renold Chains Canada, Limited
In United Kingdom: Renold Chains, Limited
Distributors in Principal Cities

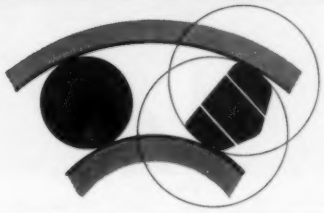


FORMSPRAG CLUTCHES

World's Largest Exclusive Manufacturer of Over-running Clutches

HOW IT WORKS

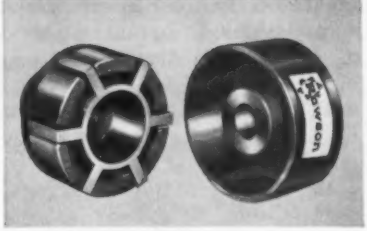




An expanding coil spring keeps the sprags in light contact with both inner and outer races. There is thus no lost motion, the driving torque being instantaneously transmitted between races. The Formsprag Clutch is so designed that it will transmit a greater torque in relation to its size and weight, than any other comparable type of clutch . . . specify Formsprag on over-running, back-stopping and indexing applications.

Forcing a ball or roller into a curved, wedged space is an old over-running clutch principle. The sprag is, in effect, a "roller" of increased diameter with greater contact surface in a given annular space. Formsprag Clutches engage at constantly changing contact points. Clutch life is prolonged and backlash eliminated. Also, with the inclined surfaces discarded, more sprags can be inserted to increase torque capacity.

FOR NO-LOAD STARTS



Specify Rawson Automatic Centrifugal Clutches and Clutch Couplings. They protect the prime mover, eliminate need for costly reduced voltage starting equipment, permit use of less expensive smaller motors to start high inertia loads. Completely automatic and entirely mechanical in operation. Rawson clutches provide full overload protection, never need adjustment. Write for the new Rawson Clutch Catalog now.

For more information, circle No. 42 on the Reader Service Card

MEN

of the power transmission industry

Maurey Mfg. Appoints Conroy and Taylor

CONROY TAYLOR

Appointment of Robert L. Conroy and Joseph F. Taylor to executive posts is announced by the Maurey Manufacturing Corp. of Chicago, Ill. Conroy is now power transmission sales representative in Wisconsin. Taylor was appointed to an executive position in Maurey's expanding V-Drive Sales Division. Conroy was formerly Branch Manager of the Power Transmission Specialty Co. As Maurey's Wisconsin representative, his headquarters are at Eagle. Taylor brings to Maurey, 28 years of V-belt drive experience with the Industrial Products Division of the Goodyear Tire and Rubber Co.

Hewitt-Robbins Elects Gay

E. Laurence Gay has been elected secretary-treasurer of Hewitt-Robins, Inc., Stamford, Conn. He was formerly associated with the law firm of Debevoise, Plimpton & McLean, New York, and from 1952 to 1953 he was a member of the legal staff of the United States High Commissioner to Germany, participating in negotiations with representatives of the English, French and German governments on various international commissions. Prior to the period of government service, he had been associated for three years with the New York law firm of Root, Ballantine, Harlan, Busby and Palmer.

Mr. Gay, a native of Bridgeport, Conn., attended Georgia School of Technology.

ACME Chain Appoints Taylor

C. K. Taylor is now Michigan representative of the ACME Chain Corp. of Holyoke, Mass., with headquarters in the Fisher Bldg., Detroit.

Taylor, affiliated with Lor-

TAYLOR

Mac Associates, has wide experience in the power transmission field, including test engineering at General Motors' proving grounds, sales engineering for Curtiss-Wright Clutch Division, and sales with Morse Chain Co.

Taylor is a graduate of the Detroit Institute of Technology. Newly elected fellow of the

ASME Elects Fearnside

FEARNSIDE

American Society of Mechanical Engineers, Thomas A. Fearnside, is affiliated with the Stone and Webster Engineering Corp., of Boston, Mass.

Active in many professional groups, Fearnside is a former Chairman of the Boston ASME section, Vice President of the Massachusetts Society of Professional Engineers, and is a member of the National Society of Professional Engineers.

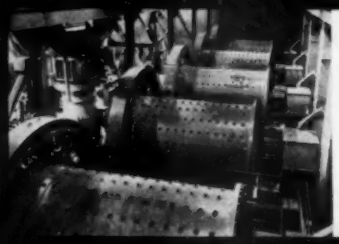
He attended Massachusetts Institute of Technology and the Harvard Business School, lives in Wellesley, Mass.

Smith Heads Crofts U.S.A.

New Chicago, Ill., manager for Crofts U.S.A., Inc., is Executive Vice President, L. F. Smith, formerly manager of the Crofts plant in Montreal, Canada.

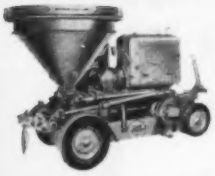
Crofts U.S.A., Inc., has launched a nation-wide drive to obtain distributors in the United States. Pioneers in the power transmission field since the turn of the century, Crofts will market a wide range of equipment in the United States. Crofts is an old, established British engineering firm with world-wide branches.

(Continued on page 20)



ALLIS-CHALMERS

TSP spur gears, spur pinions on Vibrating Screens and Ball Mills



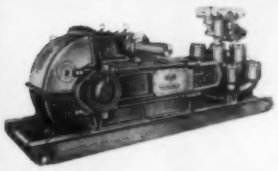
CHAIN BELT

TSP hardened products on Pumpcrete Cylinders

GENERAL AMERICAN

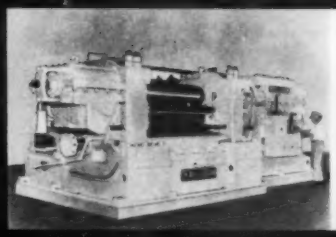
TSP trunnion rollers on Dryers

Why do these equipment builders use TSP hardened products?



WHELAND

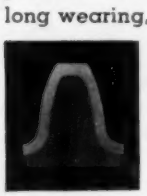
TSP gears and pinions on Slush Pumps



HALLDEN

TSP gears, pinions, racks on Flying Shears

"Tool Steel Process" hardened products when used as components, virtually eliminate replacement problems. Fewer replacement expenses, maintenance savings, reductions in costly "downtime" mean greater customer satisfaction. That's why Allis-Chalmers Manufacturing Company, Chain Belt Company, General American Transportation Corporation, The Wheland Company and The Hallden Machine Company and many more use TSP hardened products to increase the over-all quality and life of their equipment. Here are a few reasons why TSP hardened products are so desirable. They are phenomenally long wearing, a result of our special hardening process. The file hard surface to the full depth of permissible wear gives maximum life. The core, refined for toughness and ductility, gives maximum strength. Note even depth of hardness of gear tooth cross section.



TSP hardened products are made to order from your blue prints or to fit existing equipment. They include gears, pinions, track wheels, sheave wheels, axles, rolls, bushings, liners, sprockets, screw down screws and nuts, shafts, drive clutches, trunnions, hoist drums and many others. Gears are made in sizes to 90" diameter. Other products up to 22,500 lbs. They are guaranteed in writing to outlast, out-perform any competitive part in the same service.

Write for free Bulletin 352, "Why Are They Called TSP Products"

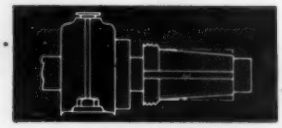
THE TOOL STEEL GEAR AND PINION CO.
CINCINNATI 16, OHIO, U.S.A.

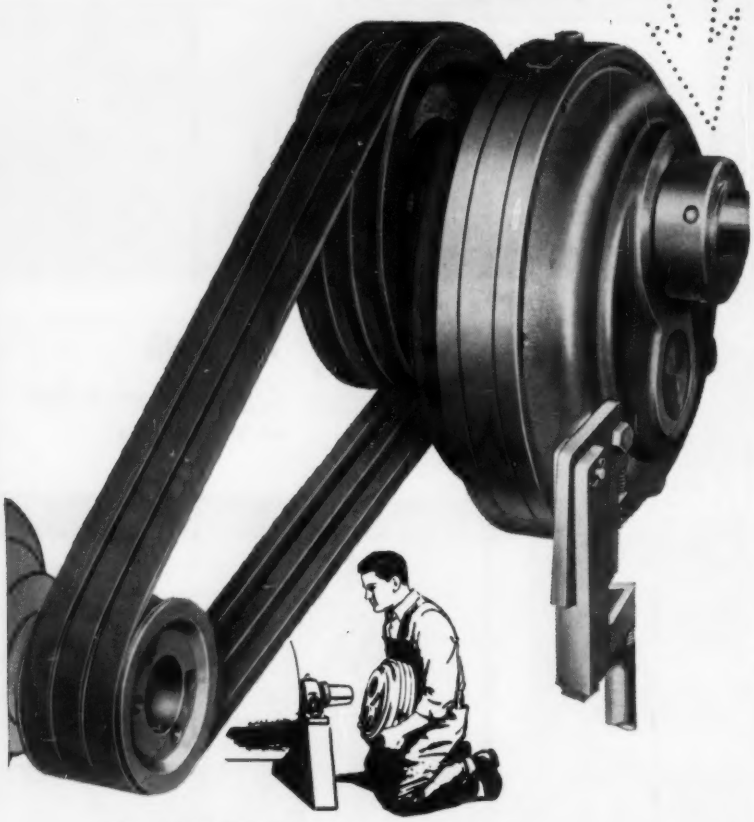
AD 1032

The Standard of Quality Since 1909 for Gears • Pinions • Rolls • Wheels and Other Hardened Products

For more information, circle No. 89 on the Reader Service Card

Easy on . . .
Easy off with American's *exclusive* Tapered-Hub Shaft-King Speed-Reduction Drives





Plus FLEXIBILITY
Any speed you need . . . by sheave substitution or adjustable speed V-belt drives

Plus SAFETY
Unique overload release and either internal or external backstop . . . for extra safety

Plus ECONOMY
Modern tear-drop contour . . . saves space and eliminates clearance problem

Write for catalog: SRD-58

The American Pulley Company
Philadelphia 29, Pennsylvania
Distributors in principal cities

For more information, circle No. 2 on the Reader Service Card

MEN *continued*

Miller Now Vice President At Gear Grinding

Promotion of Fred F. Miller to vice president in charge of engineering was announced by

the Gear Grinding Machine Company, Detroit, Mich.

Miller formerly was vice president of Detroit Bevel Gear Company, until recently a Gear Grinding subsidiary. He started with Gear Grinding in 1934.

The company manufactures automatic gear grinding machines and constant velocity universal joints.

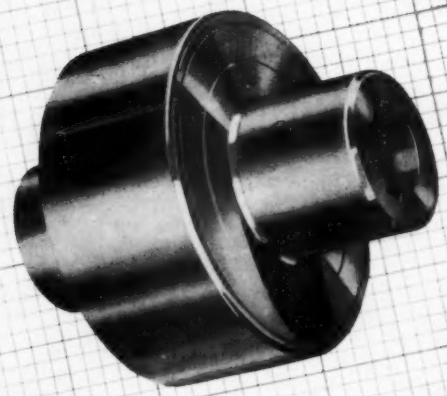
Sheehan Heads International Division

John J. Sheehan has been appointed general manager of the International Division of Hewitt-Robins, Inc., succeeding Harold Von Thaden, who is retiring after 35 years of service with the company.

Mr. Sheehan joined Hewitt-Robins in 1934 as controller of the Robins Engineers Division, New York City. He was appointed assistant to Mr. Von Thaden in 1951 and became manager of foreign operations in 1954.

In his new position he will be responsible for all of the company's international activities, consisting of an export department in New York City and subsidiaries and affiliates in Canada, Great Britain, Holland, France and South Africa.

when is a custom engineered clutch . . . a standard unit?



when you specify

CENTRIC

CENTRIFUGAL CLUTCHES AND CLUTCH-COUPLINGS

THE ONLY STOCK CLUTCHES OR CLUTCH-COUPLINGS ENGINEERED FOR SPECIFIC POWER DRIVES

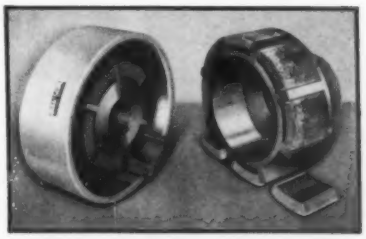
Why try to fit "shelf-item" stock clutches or clutch-couplings to your power drives—units that will not exactly meet your needs?

Centrics are assembled from expertly engineered stocked components to fulfill every required variation—"custom tailored" for every order specification.

Why install clutches or clutch-couplings that give inadequate acceleration control, torque limit and overload protection?

Only Centrics have two shoe construction (an outer and an inner) for smooth cushion starts, controlled acceleration and drive protection.

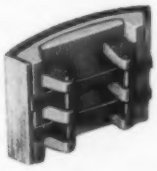
All shoe components are stocked, custom engineered to application requirements, either original or replacement.



Outer and inner shoes. Total weight of these shoes gives complete drive control.

All Centric spring-controlled units are custom designed to specific order specifications. Permanently held, self-aligning springs give longer lasting, more positive engagement control for dual drive or engine requirements.

Our long experience in specialized clutch problems enables us to fulfill your needs. Your inquiry is invited.



Shoe and spring assembly for engagement control.

CENTRIC *Clutch Company*

P. O. BOX 175 • U. S. ROUTE 9 AT MAIN STREET • WOODBRIDGE, N. J.

For more information, circle No. 29 on the Reader Service Card

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associate editor

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western editor

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art director

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distilled writing

We'd like you to meet . . .

. . . our staff. Welcome from all of us.

You are looking at the first issue of a new national magazine devoted exclusively to news and ideas about machine drives and related components.

Like you, we are not new to the power transmission field. We have been in it for many years and our parent company, The Industrial Publishing Corporation, is one of the country's leading publishers of specialized technical magazines.

So here it is. We're happy to present you with the first and only magazine of the dynamic power transmission industry. Its for you—men who are specialists in the design, application or maintenance of power transmission equipment in all types of industry.

Use this magazine like another tool in your kit. The more often you take something out of these pages, the better the tool is doing its job. Let us know how we can make it more useful to you in your work.

If you want more information about any item described in this issue, we have one of the business press' most modern reader service departments ready to serve you. However, we would like to hear your personal comments if you'd care to write us direct.

Either way, we're here to help you and hope you like the magazine.

David Cartwright

Editor

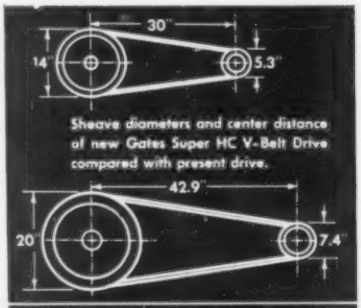
How savings multiply with new Gates Super HC V-Belt Drive

The cost of a new Gates Super HC V-Belt Drive is as much as 20% less than the cost of a drive of comparable hp using present V-belts. But the lower cost of the drive itself is only the initial *purchasing* economy.

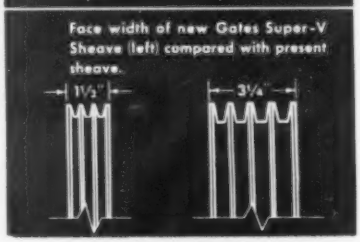
Further economies are realized on drive equipment, such as housings and bases — economies in materials, production time, shipping costs.

Shown below are space savings of a typical installation . . .

	Present Drive	Super HC Drive
DriveR Sheave	7.4	5.3
DriveN Sheave	20.0	14.0
Center Distance	42.9	30.0
Belts	4-B128	3-3V900



Sheave diameters and center distance of new Gates Super HC V-Belt Drive compared with present drive.



Face width of new Gates Super-V Sheave (left) compared with present sheave.

Super HC V-Belts also have these important Gates features...

1. **Flex-Weave Cover** (U. S. Pat. 2519590) A Gates exclusive. Makes cover more flexible . . . prolongs belt life.
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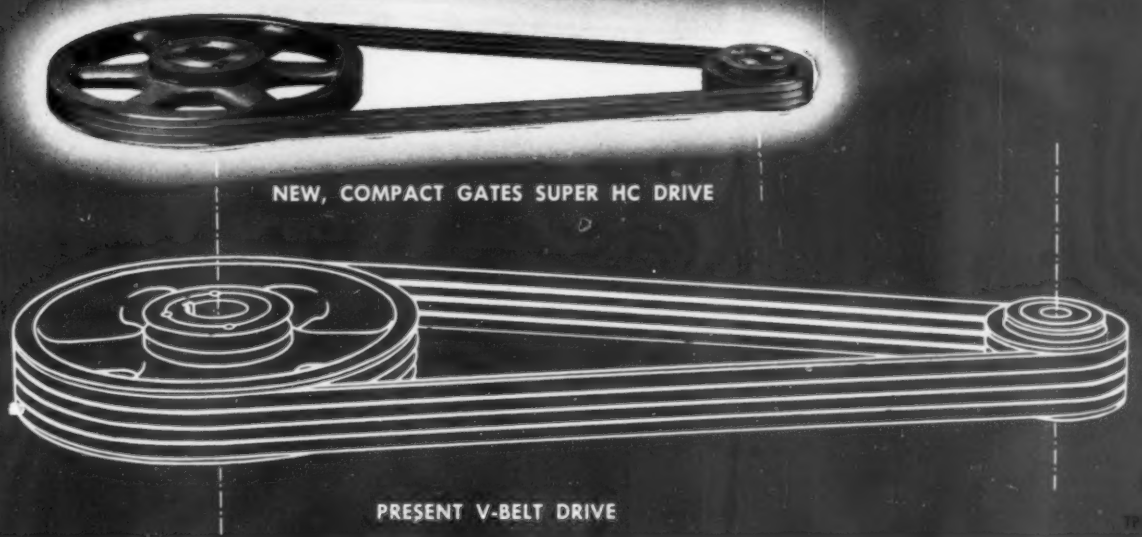
Gates

New high capacity V-belt revolutionizes drive design



Makes drives far more compact... cuts cost as much as 20%

Here is a major advance in the field of power transmission — the fully proved Gates Super HC V-Belt, developed in the world's largest belt-testing laboratories at The Gates Rubber Company.

With the Gates Super HC V-Belt, you can have the lowest-cost, lightest-weight, most compact multiple V-belt drive you can put on any machine! Sheave diameters can be reduced up to 50%, sheave widths 30% to 50%, center distances 20% and more!

On new drives, the cost of a Gates Super HC V-Belt Drive is as much as 20% less than present V-belt drives of the same hp capacity!

Learn More About The Cost-Saving Super HC Drive

For detailed information on the Super HC Drive, contact your nearby Gates distributor or Gates Office for new booklet, "*The Modern Way to Design Multiple V-Belt Drives.*" For utmost space, weight and dollar savings in new drives or drive replacements, specify Gates Super HC V-Belts and Sheaves. Stocks immediately available in principal industrial centers.

The Gates Rubber Company Denver, Colorado
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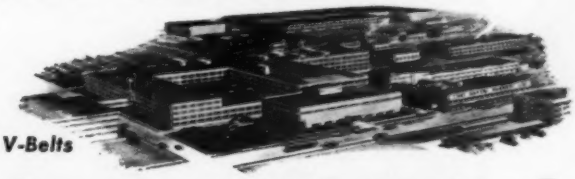
World's Largest Maker of V-Belts



Gates Super HC V-Belt Drives

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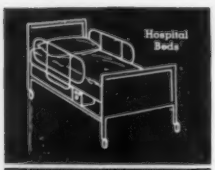

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POWER TRANSMISSION DESIGN

FEATURES

Divide driving torque evenly with

Adjustable-pitch sheaves

By **ALBERT M. ROCKWOOD**, *Battelle Memorial Institute*

TWO SETS OF ADJUSTABLE-PITCH SHEAVES, mounted on hollow shafts, transmit torque evenly to two take-off shafts. The movable flanges are displaced by hydraulic pistons in the shafts which change effective turning diameters. Belt tension is always great enough to spread the flanges and overcome friction of the pistons. The fixed flanges keep the belts perpendicular to the shafting at all ratios. Hydraulic fluid is admitted through rotary seals in the shaft-ends to control the speed ratio.

The control system consists of a hydraulic servo which regulates speed at any setting regardless of load-change. A spring makes driver-pressure control piston follow the movable flange of the driven sheave. The control valve is actuated by a double-acting ratio-control cylinder. Two pressures are applied to this cylinder: one is a function of throttle setting; the other is a function of engine speed. The combination of pump, throttle signal valve, and ratio-control cylinder is equivalent to a centrifugal governor acting on the ratio setting. As the throttle is depressed, the throttle signal valve causes pressure to act on the left end of the ratio-control cylinder.

If pressure developed is greater than pressure created by the positive displacement pump, the valve body will move to the right and expose the movable flange to sump pressure. This causes a shift toward low ratio until speed n-creases sufficiently to restore balance in the ratio-control-cylinder.

On the other hand, if the engine should speed up due to decrease in load, the positive displacement pump will speed up, creating a differential pressure on the ratio-control cylinder which will shift the valve body and open the driver-pulley pressure line to high pressure.

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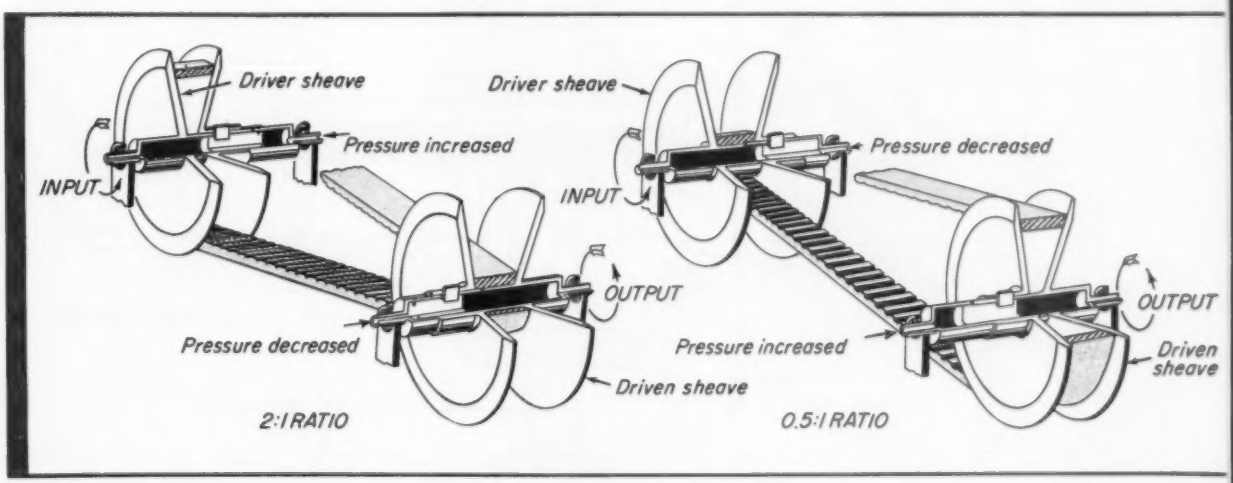


SHEAVES *continued*

Increasing pressure allows the ratio to change toward high until a balance between the pressure due to engine-speed and throttle-setting is regained.

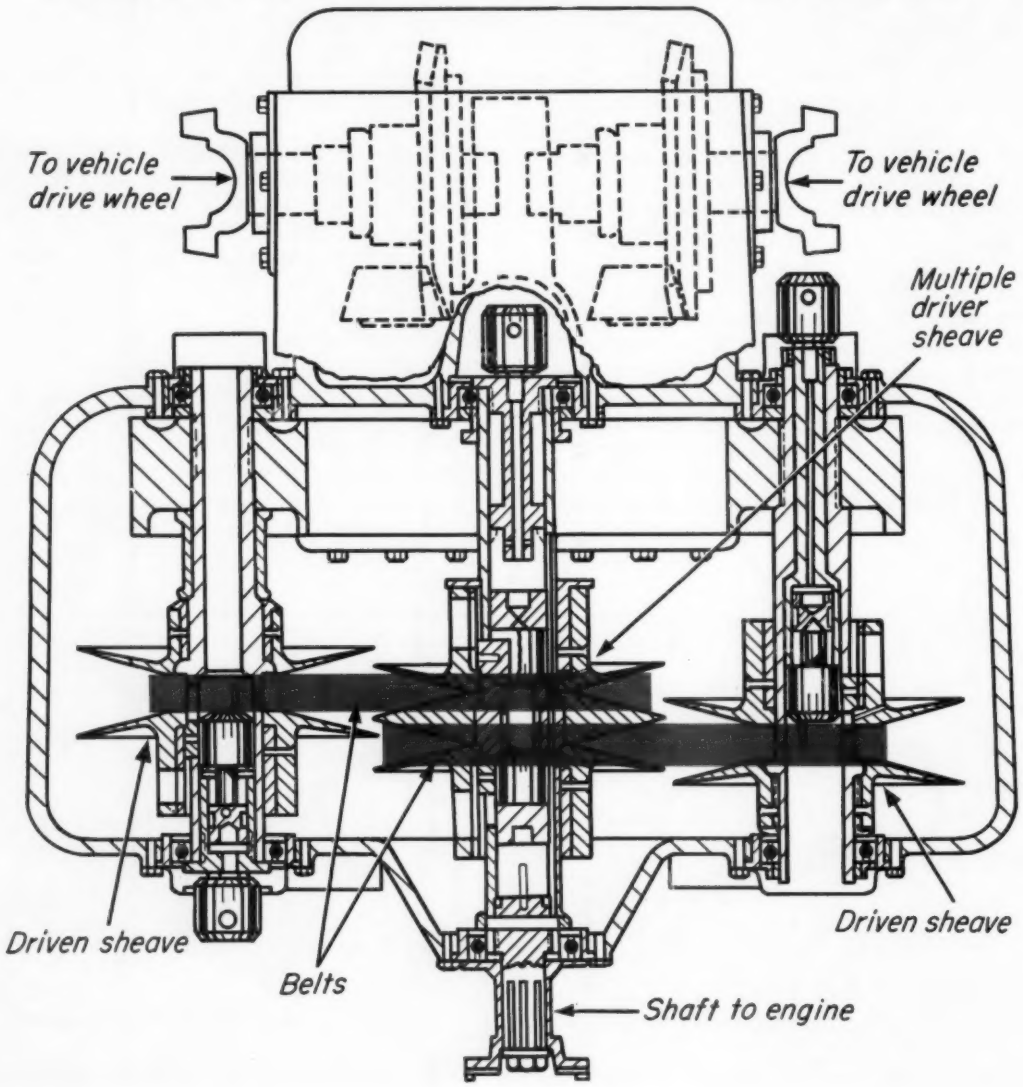
With the control-valve body fixed, any motion of the movable flange changes the piston-position which subjects the movable flange to a control-pressure, tending to restore it to original position. If the movable flange attempts to move in, due to a decrease in torque, the valve-piston uncovers the drain-port in the valve body, bleeding oil from the driver-cylinder.

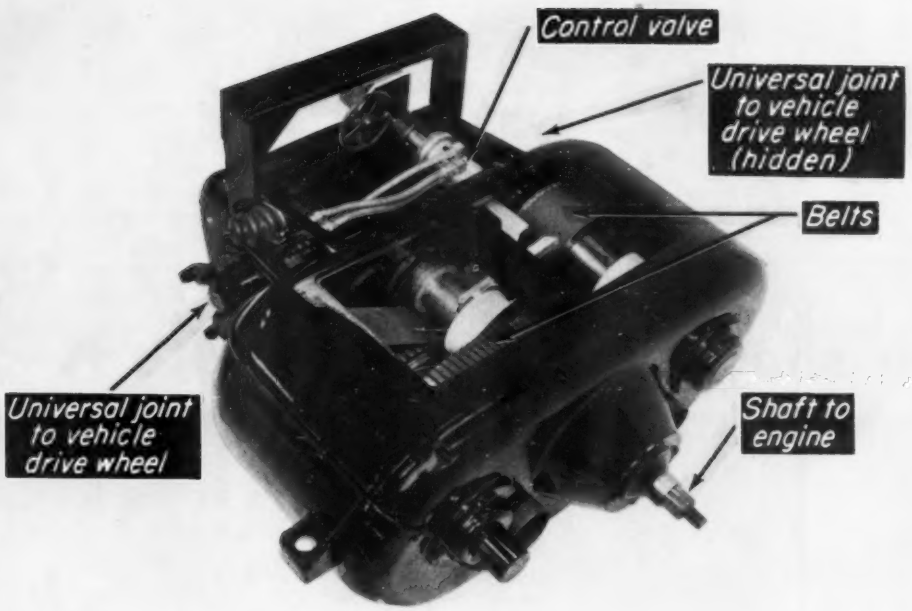
The resultant decrease in pressure on the movable flange permits the belt force to return the flange to original position. But if the movable flange moves outward, the valve-position uncovers the pressure-port, admitting high-pressure oil to the driver-sheave cylinder; locking the driver-flange regardless of the torque load imposed or pressure applied to the driven flange. Because piston movement of a few thousandths of an inch will result in a corrective pressure, the ratio can be held to close limits by locking the position of the valve-body.

Belt Slip Control

Maximum efficiency is obtained when 5% belt slip is present. This point has been defined as the optimum slip point. The optimum slip point also represents the point of maximum belt life since, for any given set of conditions, both

SPEED CHANGE is controlled hydraulically through change of belt-turning diameters.





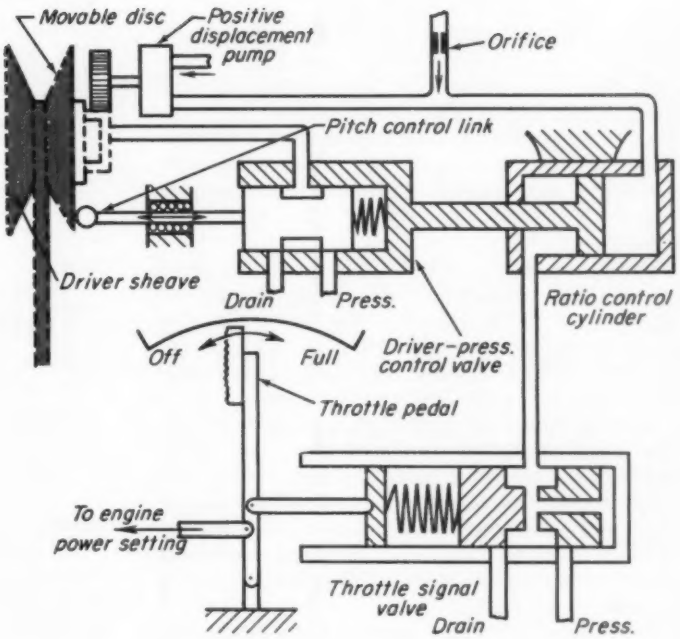
INFINITELY-VARIABLE ratio transmission uses adjustable-pitch sheaves. Belts control speed - change and perform differential action.

driver and driven-pulley pressures are at a minimum.

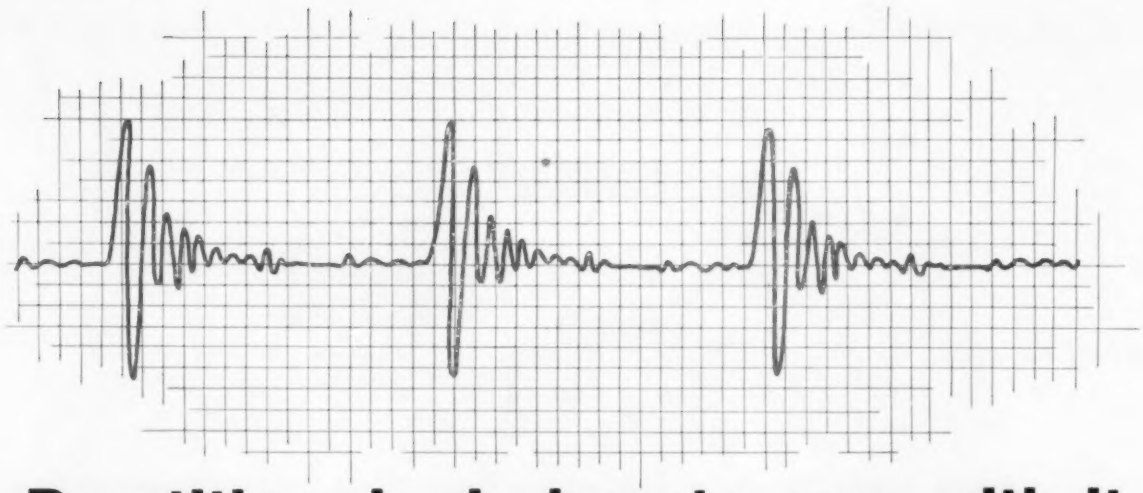
Differential Action

The independent belt drives are torque sensitive and serve as a differential. The belt drives change ratio in response to changes in torque between the two output shafts and split the torque evenly between them.

This type of drive, with the hydraulic control system, will divide torque equally among any number of driven units regardless of the change in speed of any given unit. Patents are held



DRIVER SHEAVE PISTON follows the movable driven sheave flange.



Repetitive shock: how to cope with it

By **WILLIAM A. WILLIAMS** *Consulting Editor*

REPETITIVE SHOCK IS CAUSED by recurrence of torsional peak-loads over and above average loads transmitted during continuous operation.

It differs from single-acting shock, the kind that occurs only once in a long period—such as when a machine jams. This type is not discussed in this article.

"Take It" Method

The "take it" method involves using safety factors in which gears, chain, and shafts are made larger than required to withstand peak loads. This method is used surprisingly often by designers and is a "sure thing" way of eliminating breakage problems—but it becomes very expensive and doesn't eliminate any true problems. It simply endures all the noise and vibration.

Peak loads are usually from three to ten times greater than average loads which the power drive components need be able to handle. For this reason, it is much better to use correct size parts and introduce flexible or slip members somewhere in the shaft system.

Several types of couplings are best suited. The most common are jaw type, pin type, spring grid, rubber biscuit, rubber sleeve, and bonded rubber. The best coupling for shock has the highest degree of torsional angular deflection at rated load. This coupling obviously permits the lowest amount of effective shock to pass.

Two factors govern torque or force developed by a shock peak. One is the size and speed (kinetic energy) of the rotating parts. The other is the rate of deceleration of the parts. Kinetic energy is usually fixed. Reductions in motor rotor energy can be achieved with new NEMA rerated motors. Flexible couplings soften the connection between the motor and driven equipment, thus controlling rate of motor rotor deceleration.

Anticipate the worst possible case to determine the type coupling needed.

This occurs when the shaft is brought to a dead stop instantaneously. When this happens, all the rotor energy is assumed to be absorbed by the coupling. For a single-acting coupling, the formula for the ratio of peak torque to rated motor torque is

$$\frac{T_p}{T_r} = \sqrt{\frac{KE \times 1380}{D \times T_r}}$$

where

KE=Motor Kinetic energy in ft-lb
T_r =Rated running torque of motor in-lb
D =Angular deflection of coupling at rated motor torque in degrees.

A similar formula for two couplings in series or a gear reducer with a coupling at each end is

$$\frac{T_p}{T_r} = \frac{KE \times 1380}{D \times T_r}$$

where

R=Ratio of speed reduction

Peak torque over rated torque (T_p/T_r) for a 5 hp motor for various values of coupling flexibility are shown. This indicates that one coupling has to be very soft in order to absorb all

motor energy within a torque peak of, for example, five times rated torque. But when two couplings are used, such as on each side of a gear reducer, absorption is greatly improved.

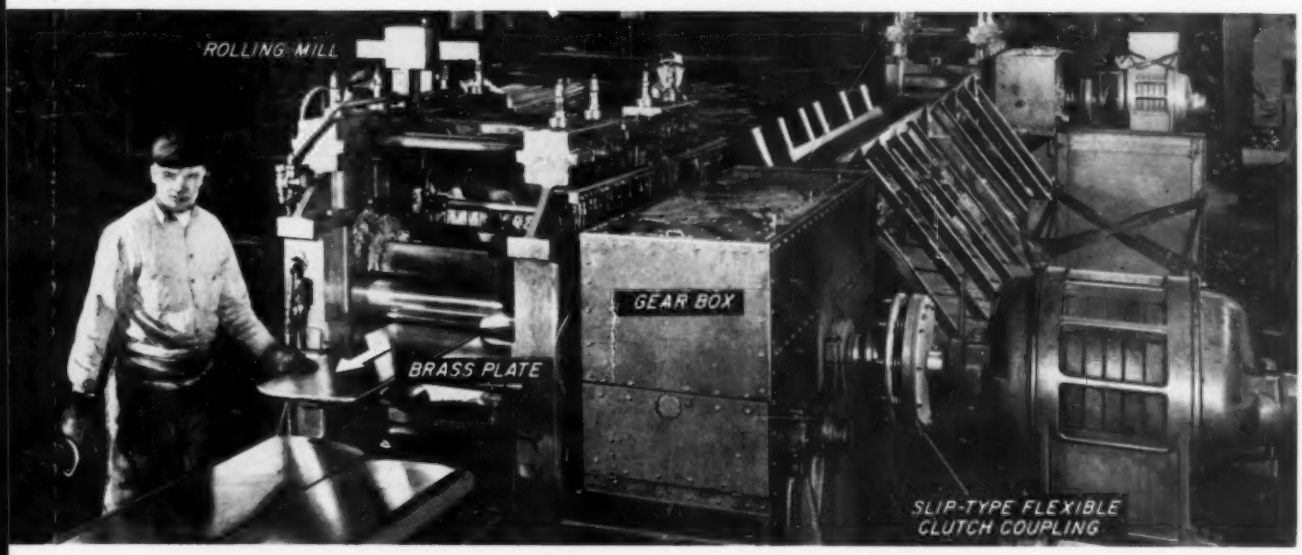
Couplings are seldom required to absorb all motor energy but sometimes it is difficult to estimate to what degree they are needed for a given application. Experience is the best indicator of this.

Slip clutch couplings eliminate most problems of motor rotor inertia. The clutch is simply adjusted to provide enough torque to start and not slip at rated load. It will slip automatically at peak loads. This means that the peak slip point of the coupling is reached every time the clutch slips. If this is two and a half times rated torque which is established to permit starting, all the components must also be engineered with a factor of 2.5.

Commercially-available slip couplings are fluid, dry fluid, centrifugal clutch, magnetic drive and slip clutches for chain drives. They improve starting conditions by permitting calculated slip but serve doubly in overcoming repetitive shock. However, they are not as easy to adjust as straight slip clutches.

VALUES OF PEAK TORQUE / RUNNING TORQUE						
Gear reducer ratio ↓	Angular deflection of coupling (deg) at rated torque of motor					
	0.5 deg	1 deg	3 deg	6 deg	9 deg	15 deg
1	70	50	29	20	17	13
5	29	20	12	8	7	5
10	21	15	9	6	5	4
20	15	11	6	4.5	4	3
30	13	9	5	4	3	2

Values of $\frac{T_P}{T_r}$ for various degrees of coupling flexibility during dead stop impact where all energy must be absorbed by coupling.



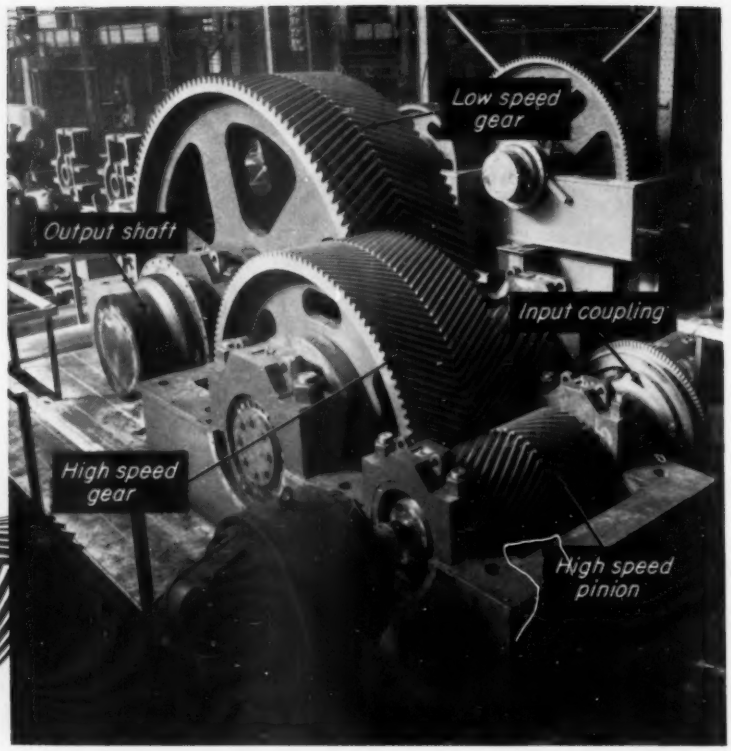
SLIP-TYPE FLEXIBLE COUPLING is shown mounted ahead of gear reducer on brass rolling mill. Coupling damps shock peak when plate enters rolls. Coupling is adjusted to slip against spring load.

Belt Drive

Flat and V-belt drives absorb repetitive shock well by stretching and slipping. They carry heavy loads and, when shock is a major problem, are better than steel cable belts, chain, gears and other direct drive methods. Stretch can be considered just as important as slip and, for this reason, they are popular on engines, beaters, hammermills, rock crushers, in oil fields, sawmills and reciprocating machinery.

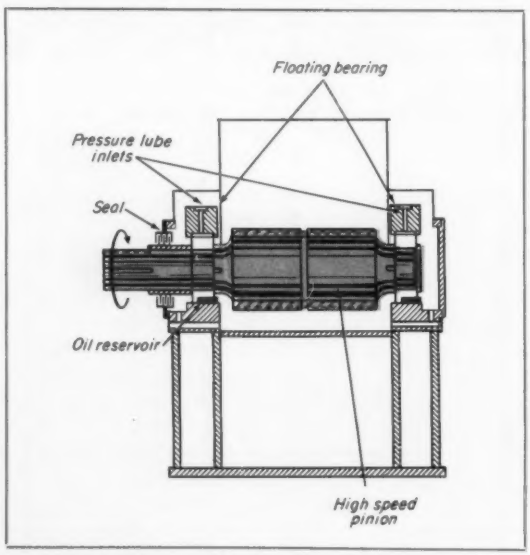
TORQUE DEFLECTION IN DEGREES FOR VARIOUS COUPLING TYPES	
Type coupling	Angular deflection at rated torque
Jaw type	1/8 to 1/2 deg
Pin type	1/4 to 1/2 deg
Spring grid	1 deg
Rubber biscuit	3 to 6 deg
Rubber sleeve	8 to 15 deg
Bonded rubber	2 to 15 deg

Double-helical gears are preferred over spur and other type gears for heavy-duty, quiet operation



Reduce noise with double helical gears

By **C. J. SCHORSCH**, *Manager of Gear Dept., United Engineering and Foundry Co.*



HIGH SPEED PINION has two floating bearings.

QUIET OPERATION of a new roughing mill is achieved with double helical gears. The gears are mounted in double reduction gear boxes and powered by heavy duty a-c mill motors. The four-stand mill reduces hot slab up to a thickness of nine inches in continuous strip mill service.

The gears are generated with a helix angle of 30 degrees. This produces a multiplicity tooth-tooth overlap which divides the load smoothly among the engaging teeth and prevents any separate tooth portion from carrying a peak load at any time. The double helix eliminates side thrust since one helix opposes the other.

The first stand is powered by a 4000 hp motor at 400 rpm which turns the gear box through a spring-type flexible coupling and shaft. Behind the gear box, a gear tooth type coupling is used to power the pinion stand which drives the rolls through two universal spindles. The pinion stand synchronizes the top and bottom rolls.

The second stand is powered by a 5000 hp a-c motor also at 600 rpm. The second double helical gear box is slightly larger, but of same design. The motor pinion is 21.6 in pd with 27 teeth; the motor gear is 109.6 in pd with 137 teeth. The main pinion is 32 in pd with 24 teeth and the main gear is 189.333 in pd with 142 teeth. This effects a total gear reduction of 30.02:1. The motor pinion and motor gear have 42 inch faces with 20 degrees pressure angle and are 30 degree double helical. The main pinion and main gear have 52 inch faces.

The high speed motor pinion is powered by the driver coupling and mounted in floating bearings. The double helix eliminates the need for thrust bearings. Forced feed lubrication is used on the bearings during operation. However, an oil reservoir at the bottom of the bearing assures sufficient lubrication during start up and after long idle periods until the forced feed system comes into operation. Labyrinth and flinger seals are used on all shaft extensions to prevent leakage and contamination. The gear teeth are spray lubricated to further effect quietness of operation.

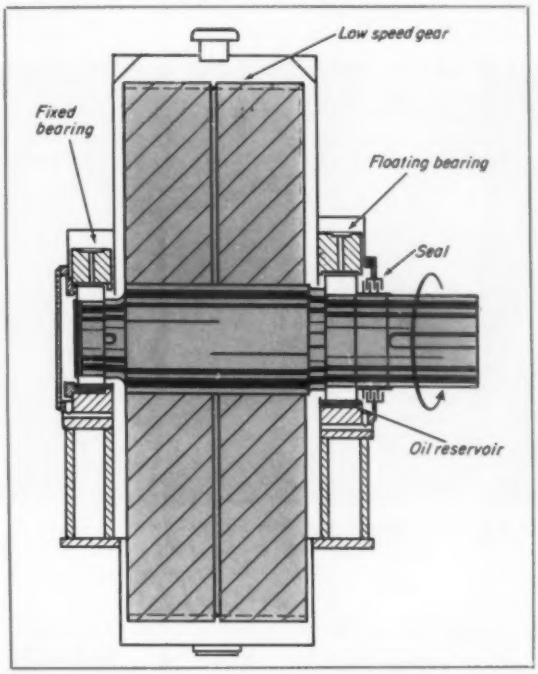
The low speed main gear has one fixed and one floating bearing. The outboard bearing is stabilized to prevent the gear from migrating.

The outboard bearing hunts as the gear turns. This permits the gear to move around for the most optimum engagement with the main pinion. Both bearings are force-feed lubricated. These bearings are mounted in an oil reservoir.

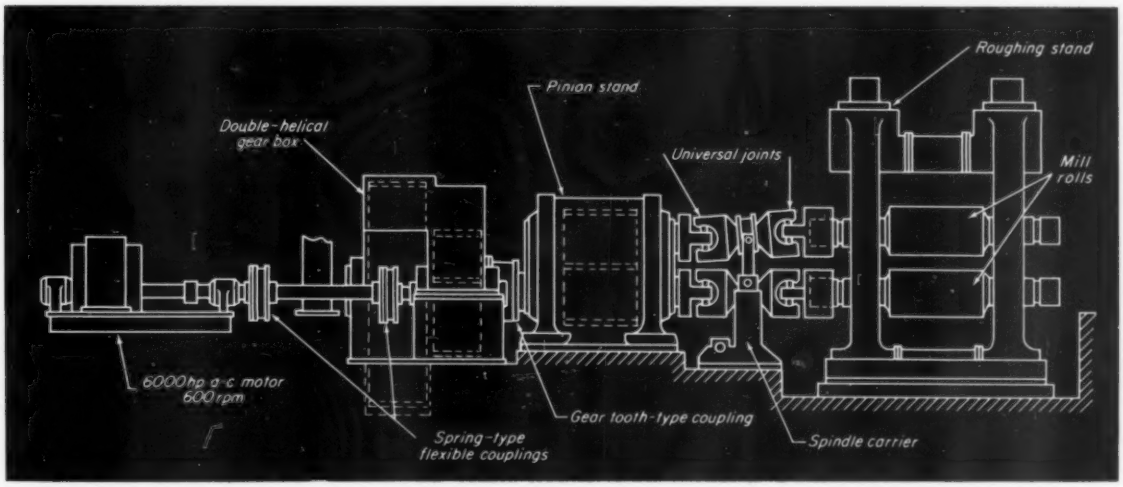
The floating aspect of the main gear is very important. If the gear is restricted there is no assurance that the teeth will carry the load evenly regardless of care in mounting and in generating. This is a circumferential situation, and not one at thrust.

The third stand is powered by a 6000 hp a-c motor at 600 rpm. The arrangement is virtually the same as the first two. The forth and final stand is also 6000 hp at 600 rpm. The mill handles hot slabs from about 4½ to 9 inches in thickness and stays on continuous duty. The slabs are reduced to strip size.

These drives are built to encounter high impact loads. The gears must be compatible with the spring type flexible couplings ahead of the gear box and the gear tooth couplings on the pinion stand side. Speed, space, alignment and reversability are also important factors.



LOW SPEED GEAR has one fixed bearing, one floating.



ONE STAND of four-stand roughing mill.

How to compute motor size and speed

Here is a method of computing duty cycles for one, or several motors, in simple applications without consulting an electrical engineer

By **RICHARD R. SELLECK** *Associate Professor, Ohio University*

RMS HORSEPOWER CALCULATION CHART

hp	hp^2	time	$hp^2 \times time$
40	1600	5	8000
100	10000	12	120000
30	900	3	2700
50	2500	10	25000
0	0	15	0
		45	155700

$$\text{rms horsepower} = \sqrt{\frac{155700}{45}} = 58.8 \text{ hp}$$

TIME, temperature, and rms horsepower

CHOOSE A MOTOR with care. It it's too small, it will overheat and have a short life. If it's too big, it will be inefficient; and if it's an a-c motor, its power-factor will also be poor, thus unduly increasing operating costs.

When a machine is designed you have to know how much power it needs, and for how long, to do its work. Designers usually plot time in seconds on the X axis and horsepower on the Y axis. This curve is called the duty cycle, and it's assumed that it will keep repeating as the machine runs. The duty cycle helps in picking a motor of the correct horsepower.

A motor's output is limited by the amount of heat it generates. It's horsepower output is about porportional to the current drawn; and heating is proportional to current squared. Thus, putting two proportions together, a motor's heating is approximately proportional to horsepower—output squared.

This conclusion leads to an important fact: If you calculate average horsepower needs from the duty-cycle curve, the answer will be wrong. What you need, in order to be right, is the average heating over the duty cycle. You can evaluate it by first squaring the ordinates of the duty cycle curve and then finding their average.

If you assume that horsepower is constant over each of the segments that comprise a duty cycle, you can simplify the calculation. To get average heating, square the horsepower for each segment and multiply each by the time it lasts, in seconds. Add them all, divide by the total duty cycle time, and take the square root of your answer. This is the horsepower you should use when you pick a motor.

This horsepower is called the "rms" (root mean square) horsepower. Its formula is

$$\text{rms hp} = \sqrt{\frac{\text{sum of } (hp^2 \times \text{time})}{\text{Running time} + \frac{\text{standstill time}}{k}}}$$

where k is a constant that cannot be disregarded, because a motor cools off much slower when standing still than when it's running idle. For open-construction motors, make **k** about three. For totally enclosed motors, such as are used in explosive atmospheres, make **k** about 6 to 8.

RMS-Horsepower Calculation

A bucket hoist needs 45 hp to close the bucket. Closing time is 5 sec. Hoisting takes 100 hp for 12 sec. Opening the bucket takes 30 hp. for 3 sec. Lowering takes 50 hp for 10 sec. The motor runs idle for 15 sec. before repeating the cycle.

Referring to the calculation chart, the rms horsepower turns out to be 58.8; therefore, specify the next larger standard-size, a 60 hp motor.

This calculation is based on the fact that the 60 hp motor will reach the same temperature

when driving the hoist as it would if operated constantly for the same time at 60 hp.

If the time units used in the example were changed to minutes, a 60 hp motor capable of delivering 100 hp for 12 min. might be chosen; but if the time unit were changed to hours, it is highly unlikely that a 60 hp motor could be obtained that would develop 100 hp for 12 hours without dangerously overheating. Reference is made to the manufacturer's data on the motor to ascertain what overload and what duration of time is permissable.

Motor Type

It's much more difficult to determine the type of motor. For drives that require variable speed, it is customary to use a d-c shunt or compound-wound motor. These types are used because it's easy to adjust their speed by either varying field current or armature current, or both. A wound-rotor, a-c induction motor could be used, but its efficiency at low speeds would be very poor. In the analysis that follows it is assumed that only constant-motor-speed drives are required. Both shunt-wound and compound-wound d-c motors are applicable for constant speed drives, with the compound-wound motor having a higher starting torque for any given size.

A-c induction motors, both single and three phase, are available in two general classifications, namely low and high slip. For a given size, the low slip motor will have a higher starting current and a lower starting torque than the high slip motor, but speed variation of the low slip motor from no-load to full-load is limited to about 3% to 5% of synchronous speed. The high slip motor will have a speed variation of from 7% to 15% of synchronous speed.

After rms horsepower has been calculated, it is necessary to refer to the speed-torque curve of the chosen motor in order to find out if the motor is capable of driving the load during the period of maximum torque as indicated by the duty cycle. Where the variations in horsepower and torque requirements of the duty cycle are not too extreme, it is usually possible to pick from the manufacturer's catalog, a motor that will have sufficient starting torque to start the load and also have a high enough peak in the speed-torque curve to drive the load through the peak of the duty cycle.

When a duty cycle is encountered with extremely high peaks, such as might be found in a drop forge or punch press, the rms horsepower equation will indicate a motor that is incapable of producing the torque required to drive the load through the high peak. The motor will need the help of a flywheel.

	a	K
VALUES OF K FOR VALUES OF a	1	1
	2.5	0.90
	5	0.80
	10	0.70
	20	0.60
	30	0.55
	50	0.48
	100 or infinity	0.42

PEAK TORQUE and kinetic energy

A flywheel and a motor must be a combination. Since a flywheel can not transfer its kinetic energy to the load without slowing down, you must pick a motor whose speed can be varied. It must be able to slow down sufficiently for the flywheel to transfer enough energy to the rotating system to carry the load past the peak in the duty cycle. The motor must also be able to accelerate the load back to its original speed, thereby restoring the kinetic energy in the flywheel before the next peak. Thus, frequency of peaks and length of duty cycle become important when a flywheel is required.

Where the load is driven through a gear train at a speed other than motor speed, and where the flywheel is running at other than motor speed, the mass polar-moment of inertia J of all rotating parts must refer to the same speed, preferably motor speed.

The kinetic energy of any rotating part in ft-lbs is equal to one half the product of the mass of the part in slugs, times the radius of gyration squared, times the angular velocity in radians per second squared.

$$KE = 1/2\ Jw^2 \text{ ft-lbs}$$

Thus, the kinetic energy given-up by a rotating mass when it slows down is equal to

$$\triangle KE = 1/2\ J\ (\triangle w)^2 \text{ ft-lbs}$$

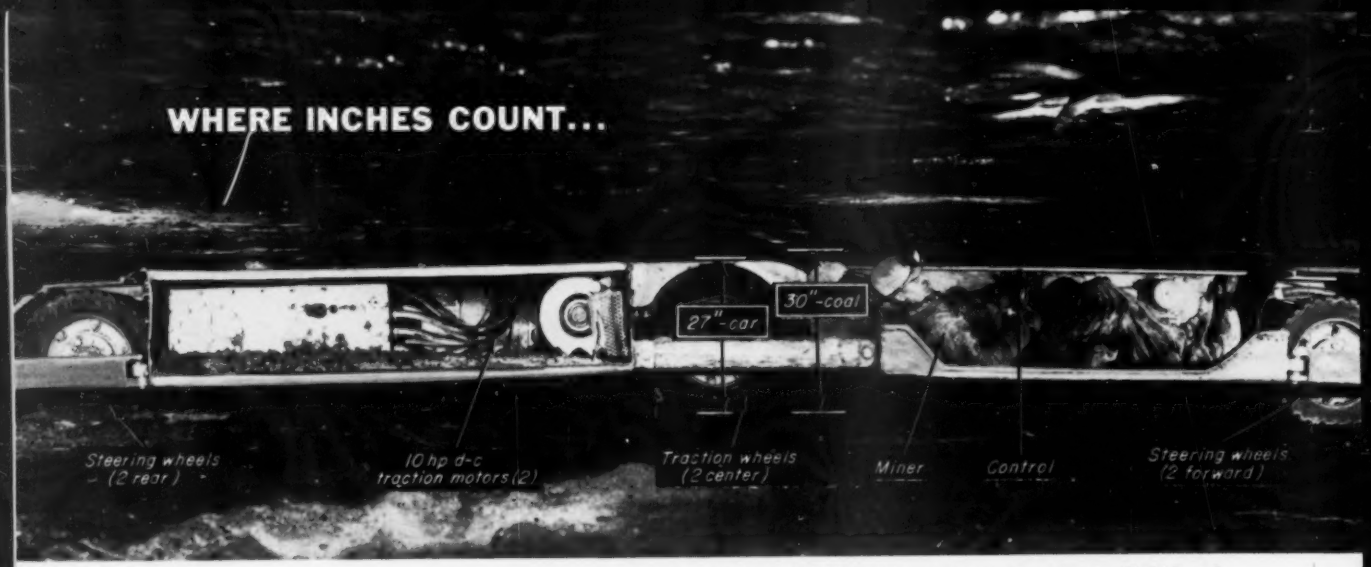
where $\triangle w$ is equal to the change in angular velocity in radians per second. Kinetic energy in ft-lbs can be converted to horse-power-seconds by multiplying by 1.818×10^{-3}.

A convenient method for calculating the amount of kinetic energy required by a flywheel is to plot the duty cycle then calculate horsepower. By drawing a horizontal line at rated motor horsepower (nearest stock motor size to rms horsepower), the peak of the duty cycle curve will fall above this rated horsepower line. The area of this peak in horsepower-seconds is

$$KE = \frac{\text{hp-sec (area of peak)} \times 10^3}{1.818} \text{ ft-lbs}$$

The relationship between J and $\triangle w$ can be worked out to produce this amount of kinetic

continued on page 82



TWO TRACTION MOTORS power center wheels. Unit bends in middle. Operator lies down while operating coal car.

Power Drives in Tight Spaces

Simplified traction and steering systems permit high payload in this newly designed coal-mine shuttle car

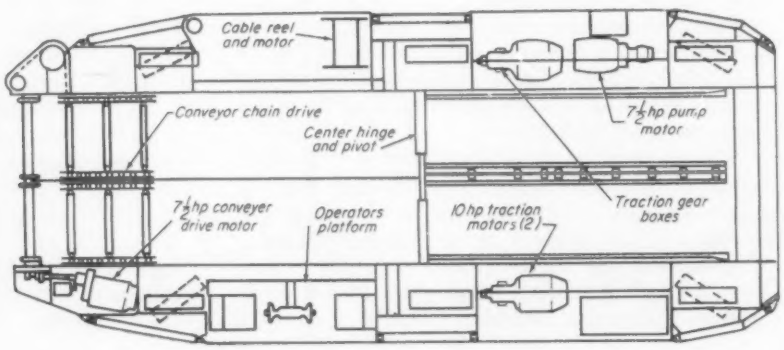
By **A. L. BARRETT** *Director, Research and Development, Joy Manufacturing Co.*

WITH SPACE A SEVERE limiting factor, power drive problems are solved with new design approaches in a new shuttle car that carries coal out of a mine.

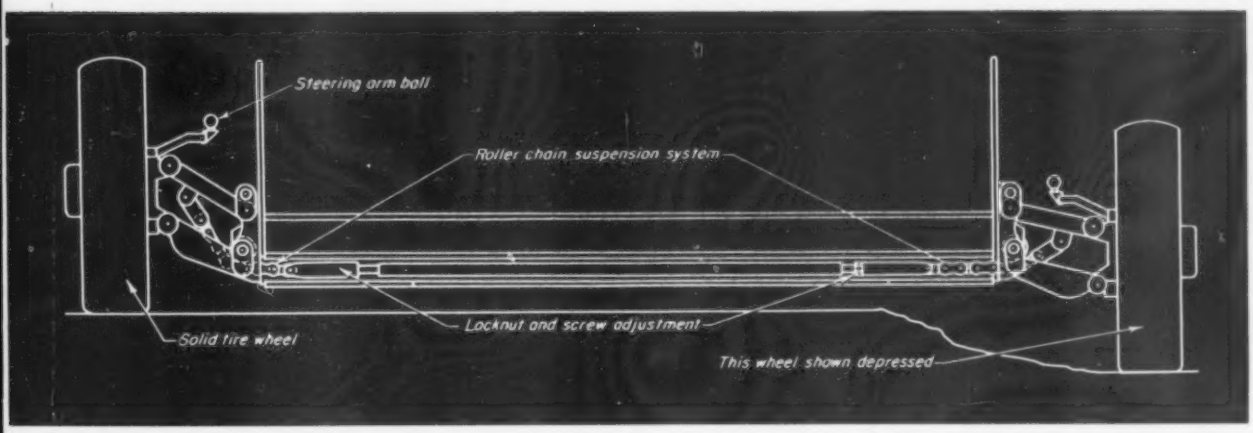
Six wheels are used; traction is applied only to the two center wheels; a roller chain suspension system replaces conventional axles; hydraulic brakes are integrated with traction gear boxes; no clutches are used; the four outboard wheels are power-steered; and a center pivot pit permits the car to bend up and down in the middle.

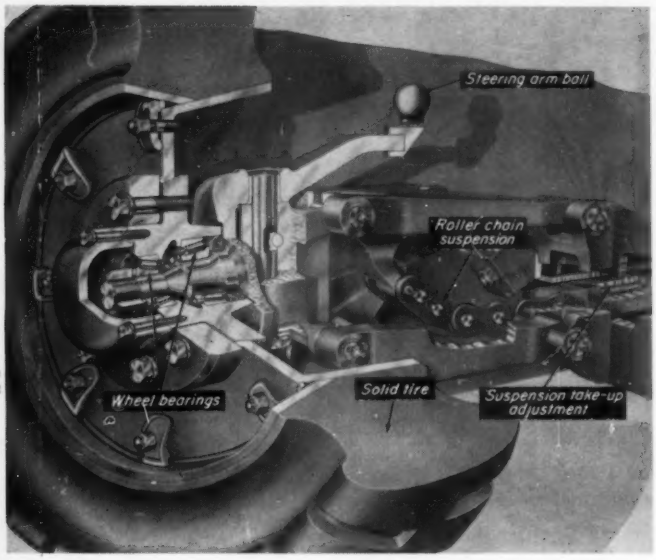
Four d-c motors do all the work. They have a combined capacity of 35 hp. Two 10 hp motors are for traction; one 7½ hp motor for the hydraulic system and power steering; and one 7½ hp motor for the conveyor drive.

Each of the two center wheels is powered separately. No cross-drive shafts nor differentials are involved. When the car goes around a corner, the series-wound motor on the outboard side speeds-up and the series-wound motor on the inboard side slows-down. This eliminates dragging wheels. Here are two features:



7½ HP PUMP MOTOR supplies hydraulic pressure for power steering of four outboard wheels. Four motors do all the work on this car that carries coal out of mines.





ROLLER CHAIN takes place of center-pivoted axle and suspension system.

AS WHEEL STRIKES bump or hole, wheels stay verticle, do not tilt.

Feature 1: roller chain replaces axle and suspension

Two steering wheels at each end are connected by roller chain. The result is similar to the wheel action achieved with a center-pivot axle.

As the wheel on one side strikes a bump or hole, the four point suspension on that side allows the wheel to move up or down.

The result: ground clearance is reduced by half the whole depth and the car body stays level. The wheel stays vertical and does not tilt, as in a conventional system. Ground clearance can be adjusted by moving one take-up nut on each chain. The articulated mid-section gives flexibility to the entire length of the car.

All wheels turn in the same direction at the same time, moving the car in a circular pattern. This turning radius is better than that made by only two wheels with equivalent wheel base. An inside turning radius of 11 ft. is achieved. When steering, no elongation nor contraction of the drive links is necessary regardless of the angle between the two ends as the car bends in the middle.

The up and down movement of the ends is compensated by movement of the ball and socket joints on the short connecting link. This is located in line with the hinge point between the two body sections.

This link coordinates the angular movement of the two cross arms which are located approximately six feet apart. Each arm rotates horizontally around a pivot point located at the conveyor's center line. The drag links are connected to each end of the cross arms and have horizontal movement.

Steering is hydraulically-powered by two identical jacks, one on each side of the car. Power steering is controlled by a steering gear and dual wheels mounted on a single shaft. This operates a hydraulic control valve through a high ratio reduction gear. The four steering wheels are inner-connected, by four drag links, to two cross-arms through the center of the frame.

These cross-arms have rotating force applied to them by the hydraulic cylinders. The amount of oil pressure in the cylinders is determined by the control valve. Thus the conventional tie rods are completely eliminated and no moving steering-parts hang below the shuttle car frame.

The two center wheels are powered directly by traction motors through worm gear boxes and chain drives. No clutches are used. The two-speed motors provide traction; the brakes stop the car.

Feature 2: hydraulic brake integrated with gear box

The brakes are made up of two basic parts: the rotating disc that turns with the gear-box shaft, and the single-piston cylinder and housing assembly. Two brake linings are located in the housing. One is fixed and the other moves with the hydraulic piston. Depression of the piston causes the movable lining to move against the disc, clamping it.

Adjustment for lining wear is automatic, since braking effort is a clamping action. The linings are easily accessible for quick replacement.

The hydraulic system is powered by the fourth 7½ hp motor. This motor drives a helical gear reducer at a ratio of 2.4:1. The reducer powers a dual pump at constant speed of 700 rpm. Braking action is performed by the miner's foot pedal that operates master cylinders. This cylinder is connected to the brake pedals by mechanical linkages. When the pedal is released, a slight wobble of the rotating disc causes the piston to back off.

Reel Mechanism

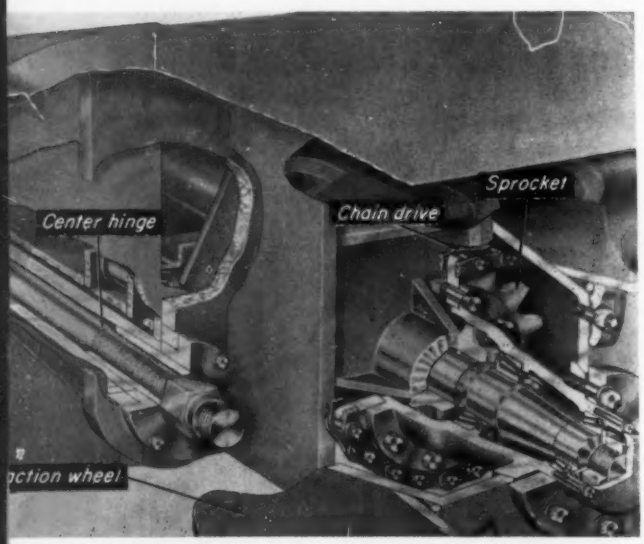
The cable reel mechanism is also automatic. The reel plays out the d-c cable. When the car is standing still or winding cable, the system is on high torque. When unwinding, the system goes into low torque. When the car comes to a standstill or starts to pick up cable, it automatically goes back to high torque.

One section of a tandem hydraulic pump powered by a 7½ hp d-c motor supplies pressure for this system. This pump section can be adjusted to deliver high pressure at 450 psi or low pressure at 150 psi as required.

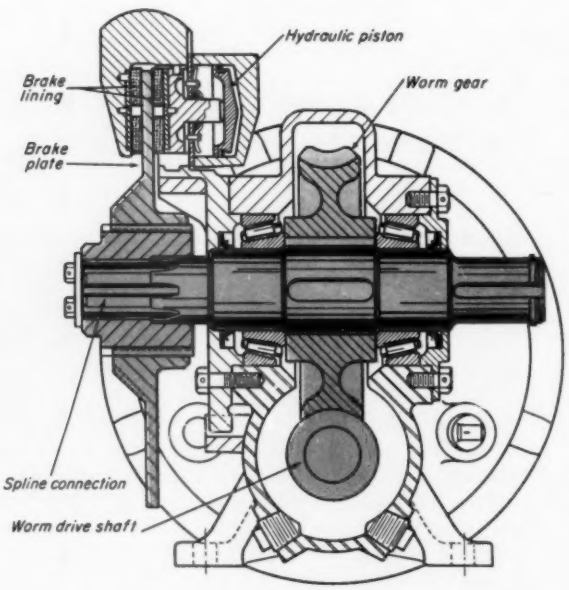
The reel is driven from the hydraulic motor by a roller chain operating over two sprockets. Rotation direction depends on whether the car is winding or unwinding cable. A large drive sprocket is fastened to the cable-reel assembly. The cable-reel is supported by two tapered roller bearings on the reel-carrier. A sprocket and cap assembly bolted to the cable-reel drives a spooling device by a second roller chain.

This device consists of a spooling shaft on which a spooling frame is mounted. The frame walks back and forth across the spooling shaft as the cable-reel rotates, paying cable onto the reel in much the same manner as a level winding fishing-reel, thus causing the cable to wind evenly on the reel.

And so we have described details of an operation in which a hydraulic brake has been successfully integrated with a gear box to produce an efficiently operating unit. Results have proven that the predictions of the designers who first conceived this application, were based on the soundest reasoning and most careful calculation.



TWO-SPEED d-c traction motor powers wheel directly through roller chain. No clutch is used.



HYDRAULIC BRAKE controls stopping.

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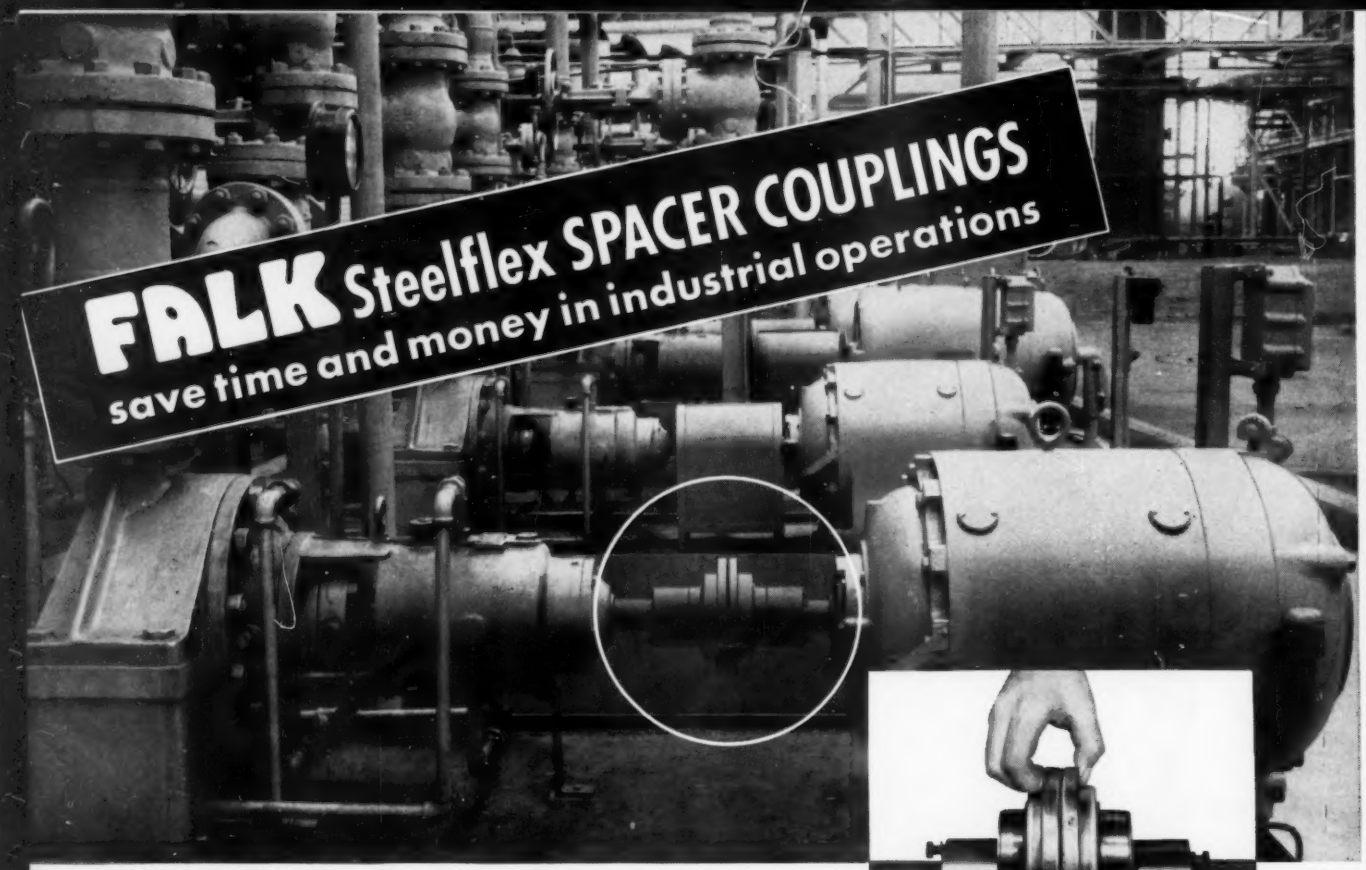

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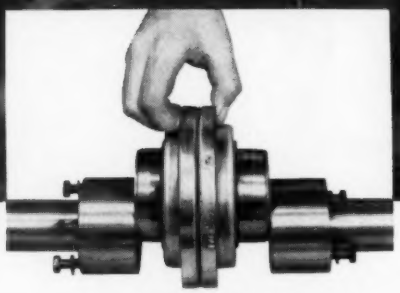

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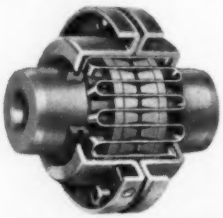

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IDEAS

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Secondary belt eliminates auxiliary motor

ADDING A TIMING BELT to the main drive shaft of a bag-filling machine synchronizes two operations and eliminates a motor.

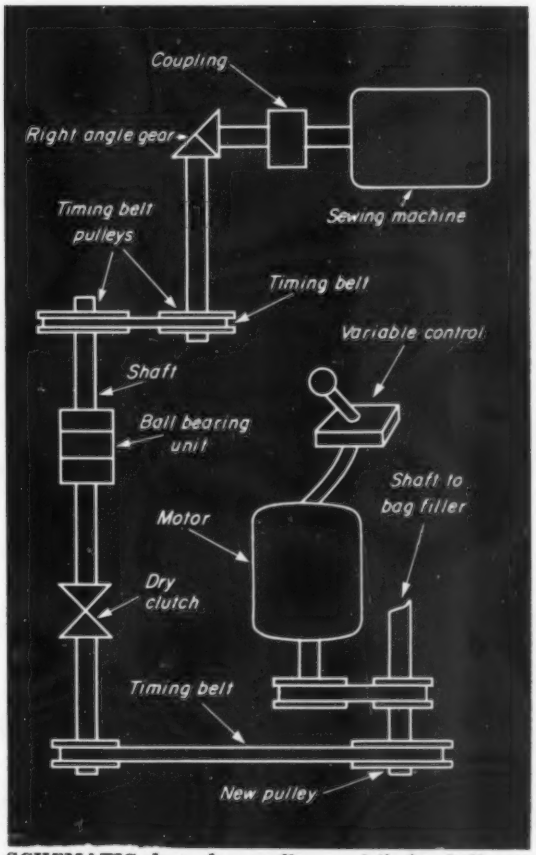
The main motor does not use more power. The belt makes the bag-filling and sewing operations 15% more efficient.

The timing belt powers an intermediate gear-reducer and line-shaft. The sewing-head is driven through a dry clutch incorporated on the shaft. When the operator engages the clutch, the sewing machine operates at the same speed as the bag-filler.

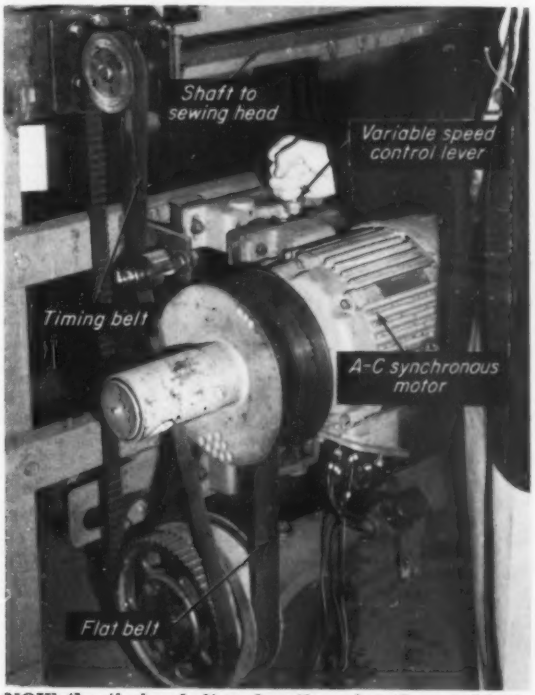
The earlier model had separate motors, could not coordinate bag-filling with sewing.

The operator had to make adjustments by hand until both speeds synchronized to 60 cycle power. This worked well during continuous running but went out of time when starting and stopping. Needles broke, bags tore.

The new synchronized unit is the development of the Consolidated Packaging Machinery Corp., Buffalo, N. Y.

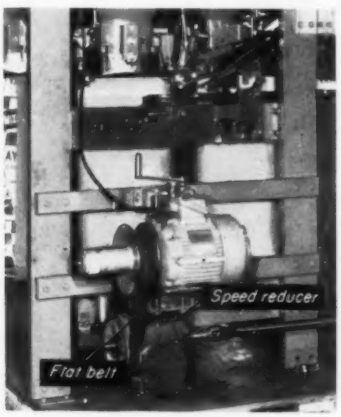


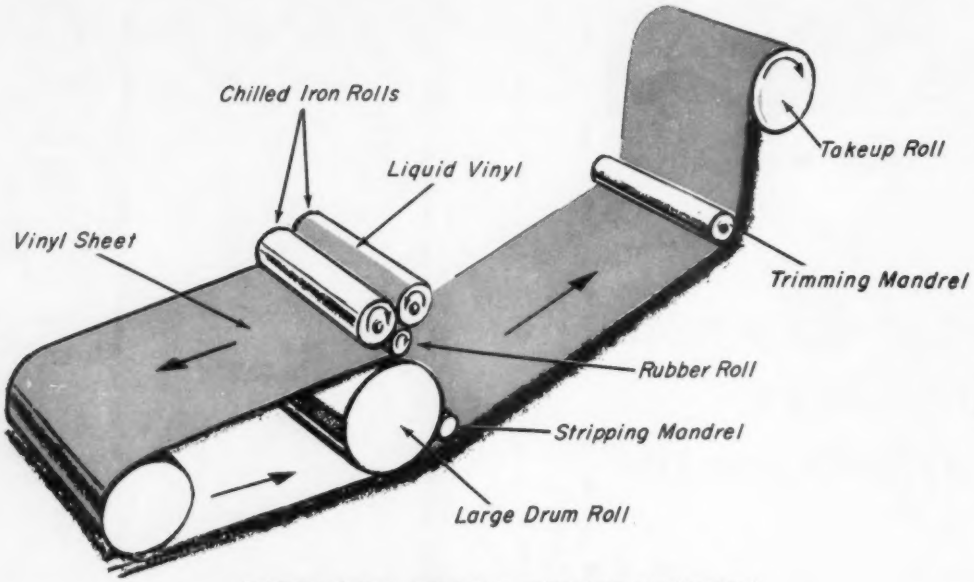
SCHEMATIC shows how pulleys and timing belts are able to transmit power where it is needed.



NOW the timing belt and pulley give the machine's operator completely synchronized control.

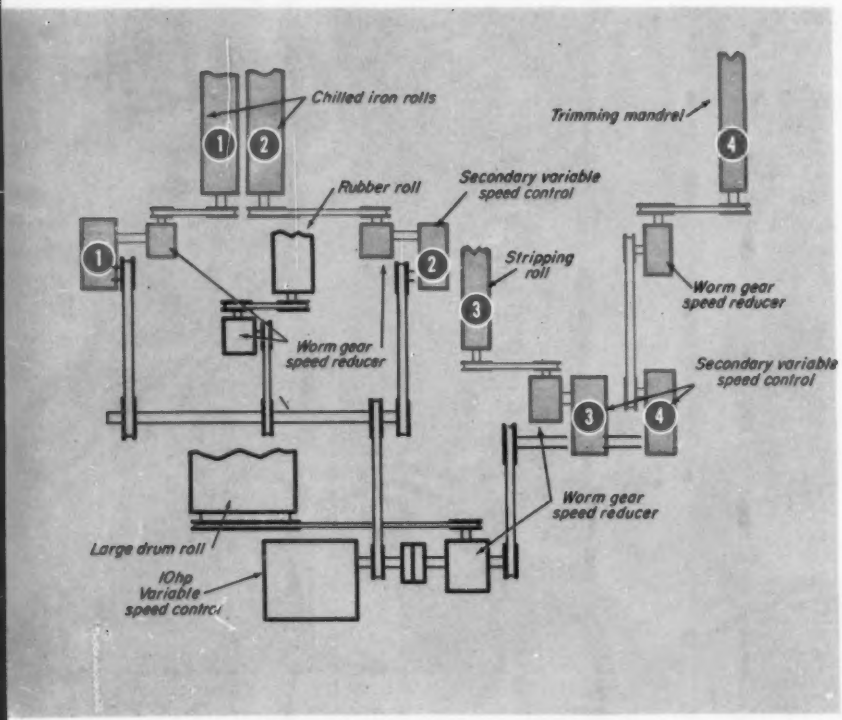
BEFORE, this motor powered only the bag-filling unit of the machine.





LIQUID VINYL RESIN is rolled into plastic sheet.

Variable-variable drive provides secondary synchronization



A 10-HP VARIABLE speed drive coupled to four secondary variable speed drives give minute speed control to a liquid vinyl resin process machine.

When primary speed is changed, all drum and roll speeds also change but remain in a fixed ratio. Range of speed change is from 25 to 100 surface fpm of the stainless steel belt which conveys the process-sheet.

Power is taken off the main shaft by a toothed timing belt. Another timing belt drives the rubber transfer roll. Speed of this roll is always constant in relation to speed of the large drum carrying the stainless belt. Applicator roll speed must be variable and independent of the transfer roll.

A double-enveloping worm gear reducer and variable speed drive are coupled between this roll and the line shaft by a tim-

ing belt. The metering roll is further variable in relation to the applicator roll. Its drive system is a duplicate of the applicator roll drive.

The entire coating head operates whenever the machine is running. However, the stainless belt can be stopped by declutching so the coating head is designed to be pivoted slightly to take it out of contact with the belt.

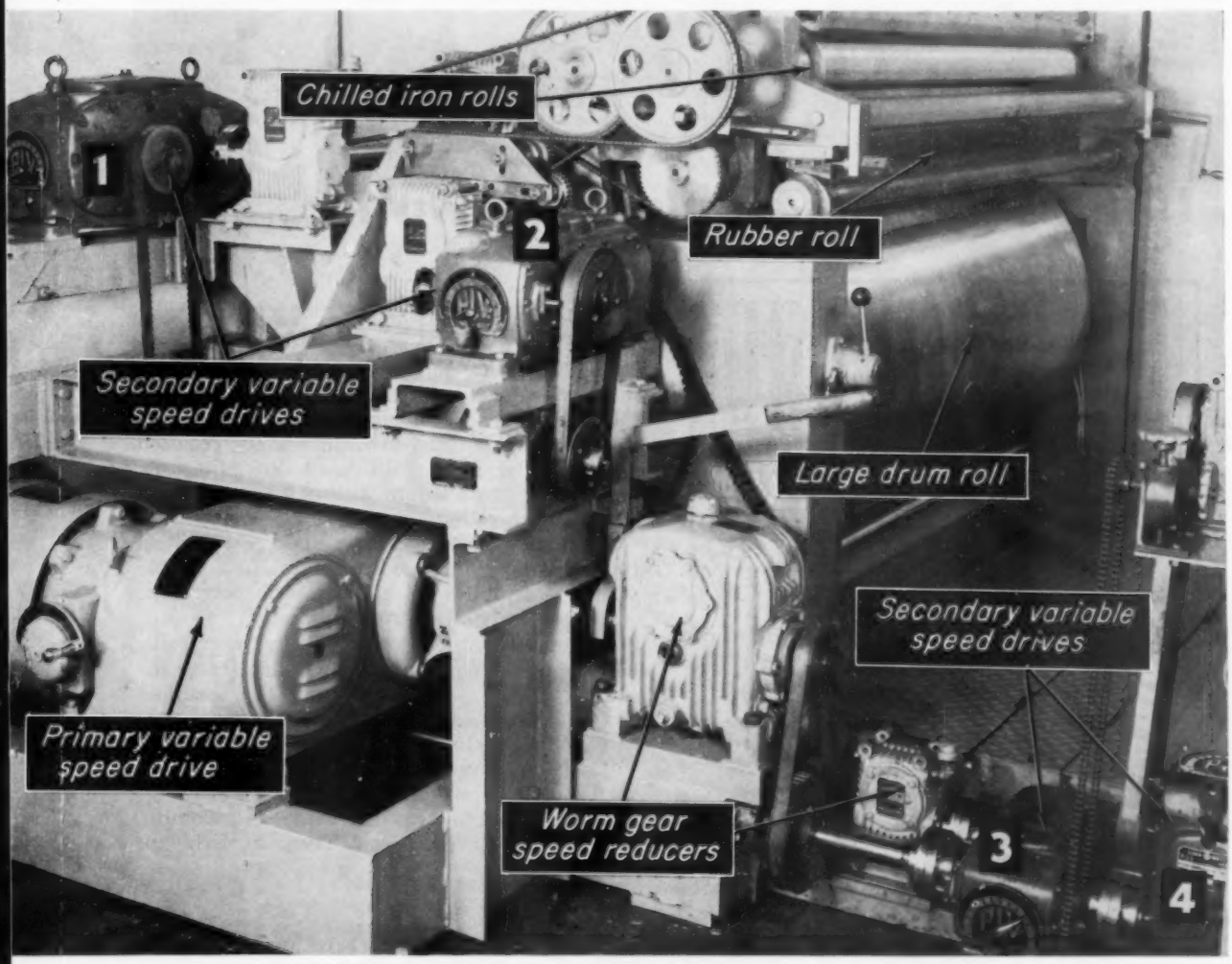
A clutch is also coupled to the coating head drive for powering a speed reducer with a double extended input shaft. The output shaft of this reducer drives the main drum and steel belt. The other end of the input shaft is connected by timing belt to another line shaft with the two remaining variable speed units.

One unit is coupled to a worm gear speed reducer which is chained to the stripping mandrel that removes the plastic sheet from the belt at the bottom of the drum. Adjustable speed is required since the thickness of the plastic, composition, and speed of casting are never exactly the same.

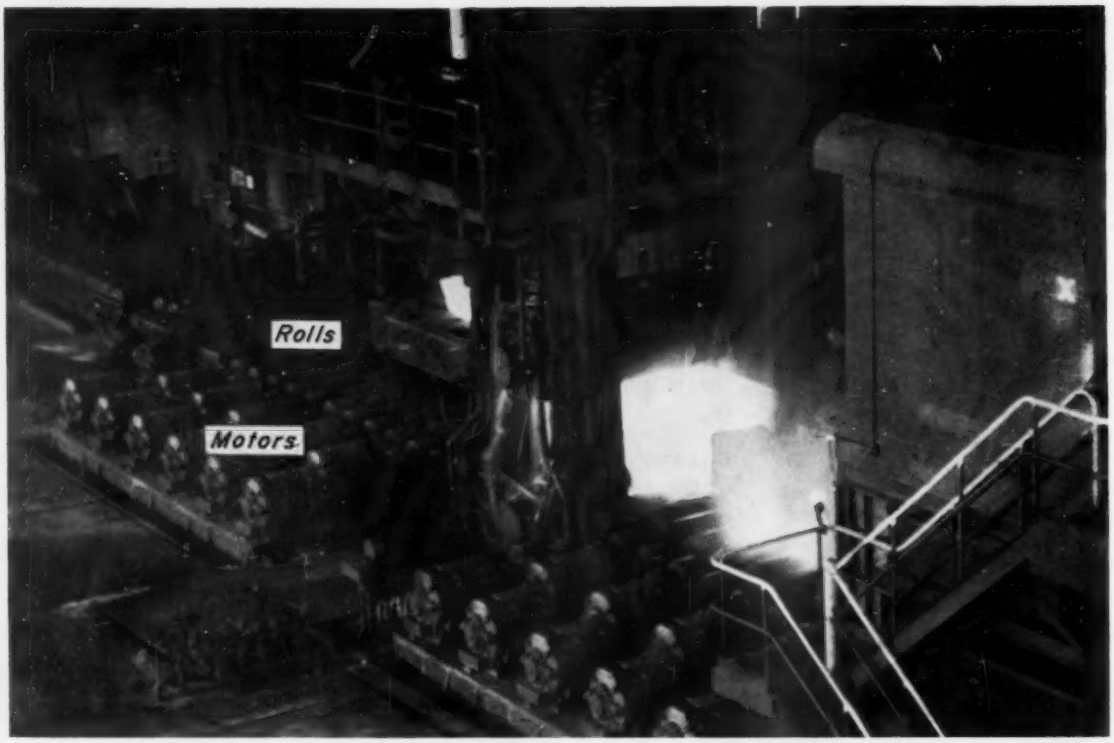
Directly coupled to the second shaft is another variable speed drive connected by timing belt to still another speed reducer, which is chained to the trimming mandrel. This mandrel is also subject to varying conditions which must be locally controlled during operation.

The vinyl resin to be cast is placed in a reservoir formed by two chilled iron rolls. Regulation of the amount of resin used is dependent on the speed of the rolls and the width of the gap between them.

Designed by Process Machinery Division, John Waldron Corporation, New Brunswick, N. J., this machine operates virtually free of noise.

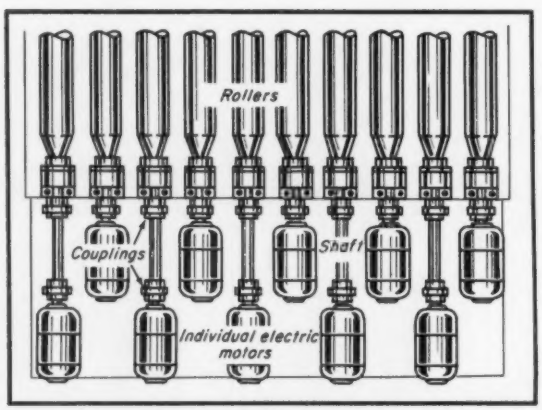


SIX SPEED REDUCERS, five variable speed drives, one motor, many belts, chain and two line shafts are powered by single 10hp motor. Four secondary speed controls give machine minute flexibility.



INDIVIDUALLY-DRIVEN ROLLERS. Motors drive rollers on blooming mill, cut downtime. Glowing, 18-ton ingot is on rollers.

Unitized drives eliminate downtime



NEW method: motors in staggered position drive rollers direct.

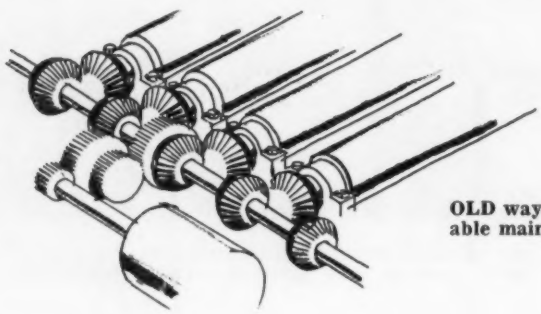
TWENTY-TWO, 18 hp, 250 volt, shunt-wound motors replace a single drive shaft with 22 take-off gears on a blooming mill. Motors drive rolls directly, cutting downtime which resulted when one or more rolls went out of action. Motors are installed in alternating position—one for each roller.

Previously, a 22-foot long shaft powered all rolls driven by four 100 hp motors. Now, double piloted assemblies alternating with couplings, at 91 rpm, drive each roller directly.

Eighteen-ton ingots move over these rolls. Motors must be large enough to accelerate ingots at a rate limited only by the assumed coefficient of friction of 0.25. To accelerate one-half an ingot, plus one roll, takes 4030 ft-lb.

Motors are equipped with thermostats which open at 115°C and close at 95°C.

This installation was made in the blooming mill at McLouth Steel Corp., Trenton, Michigan.



OLD way: 22 bevel gears powered by single drive, caused considerable maintenance when one roller went out of action.

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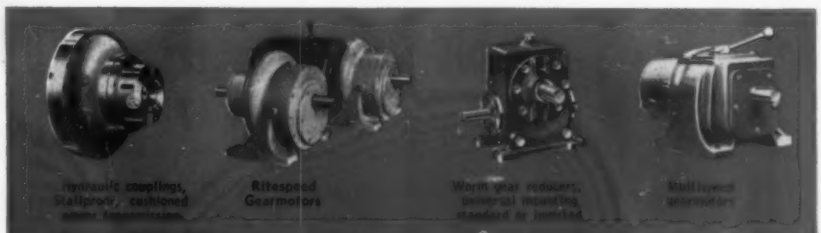

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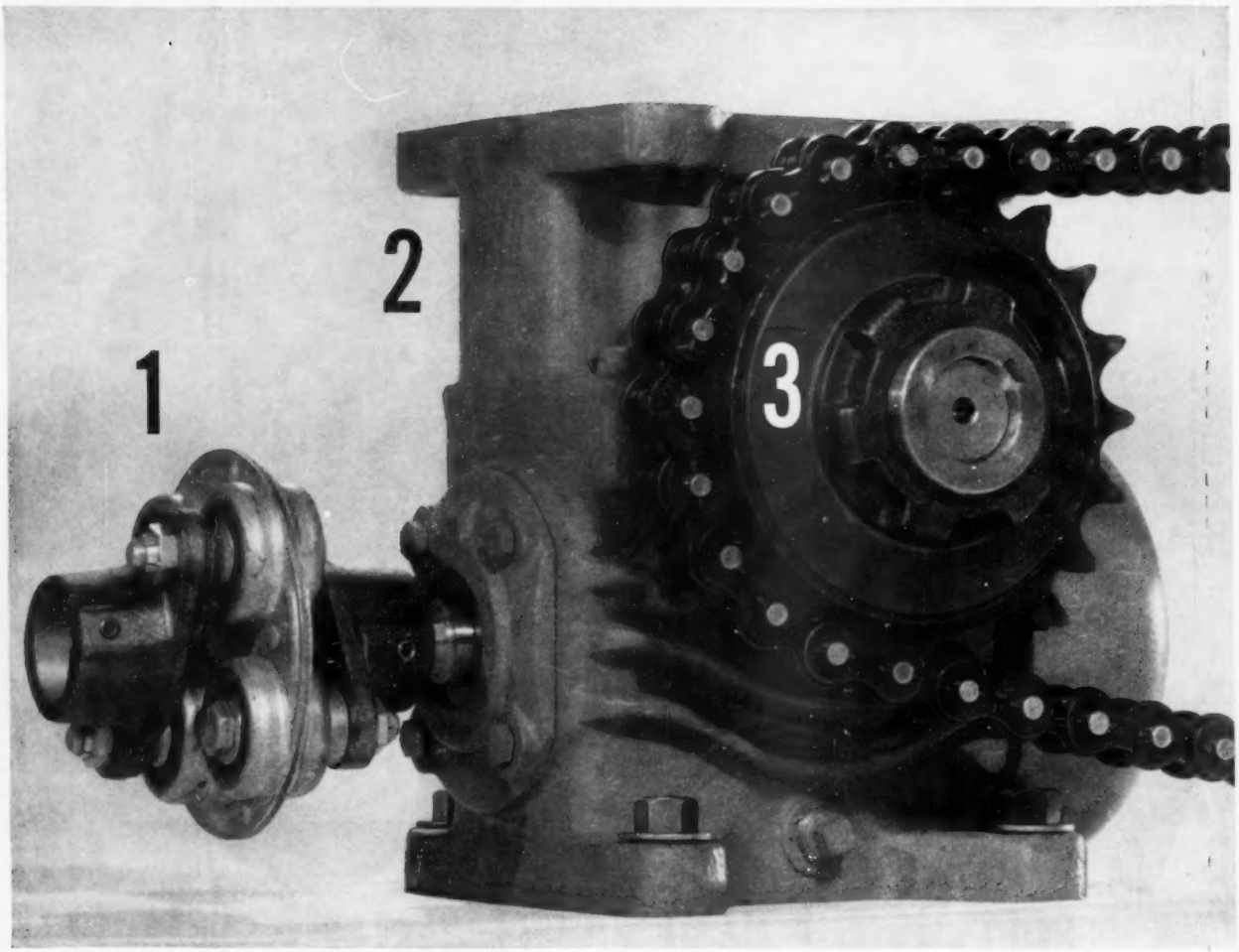

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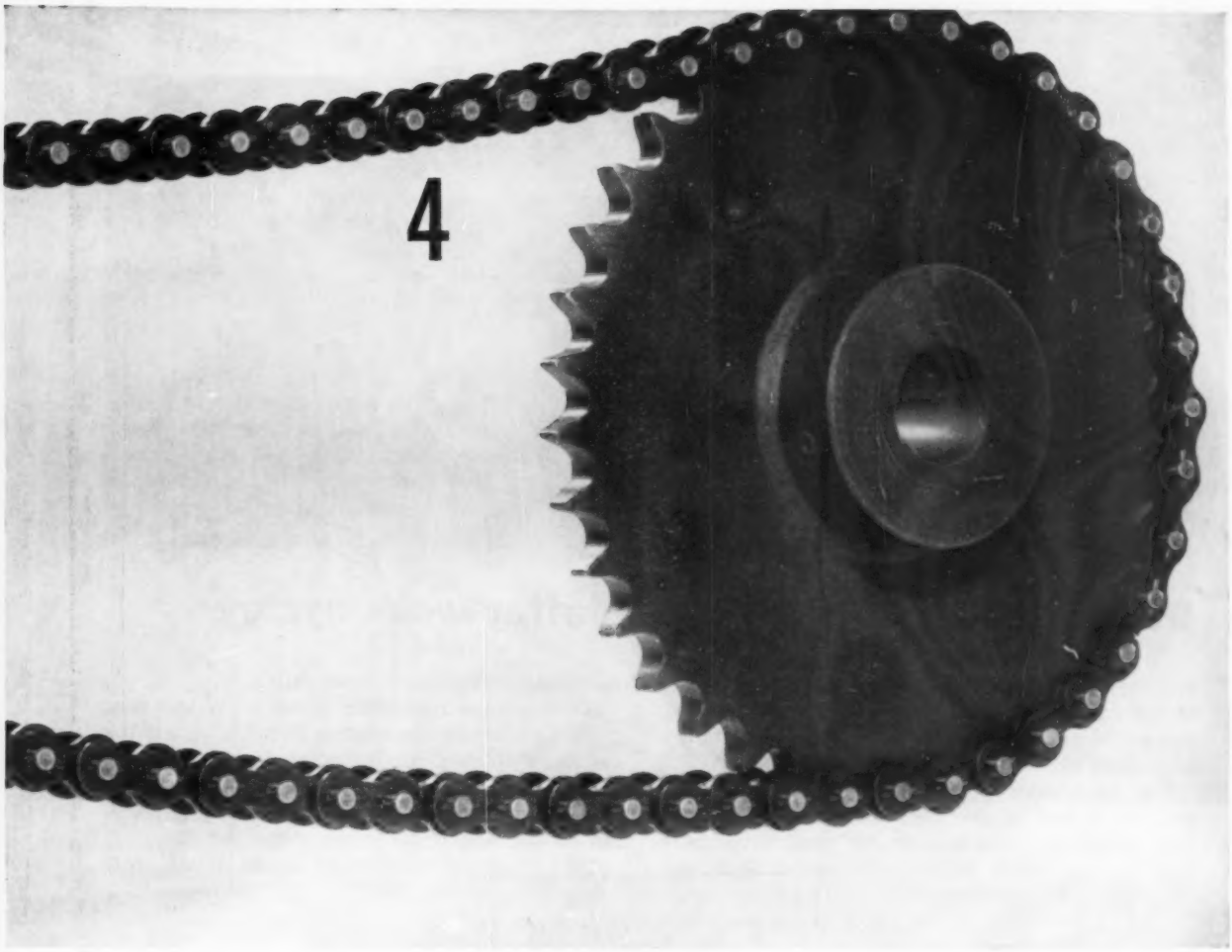

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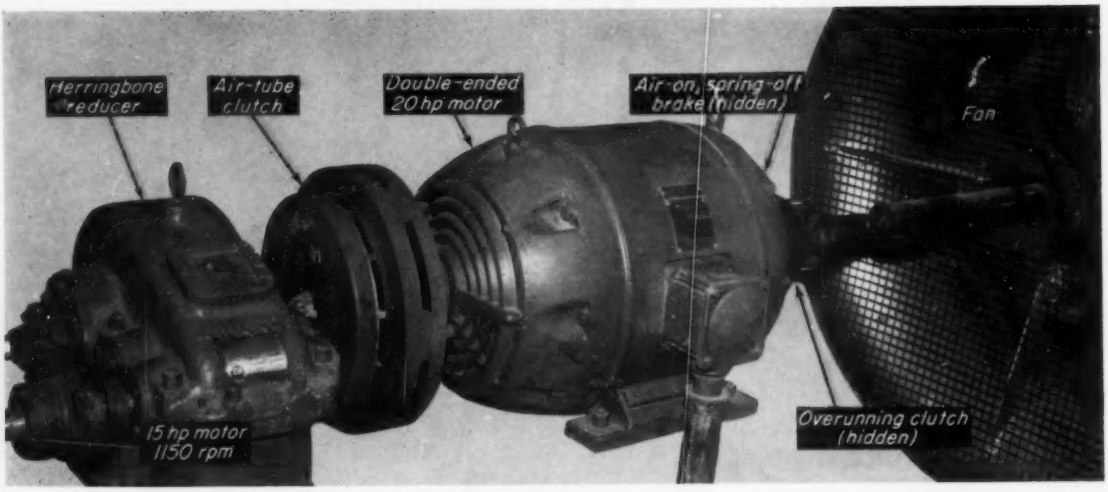


Double-ended motor cuts in after work cycle

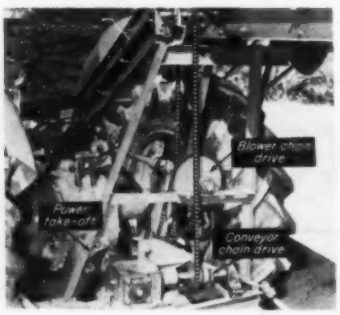
A 20 HP MOTOR with a thru-shaft, cuts in at the end of work cycles on an aircraft engine dynamometer. A 15 hp motor powers the motor as a shaft and the entire gear train.

The unit develops 1000 ft-lbs of torque for starting a test engine and slips when the engine backfires or when an oil leak develops. Torque adjustment between 700 and 1100 ft-lbs is possible for different type tests by varying air pressure to an air-tube clutch.

An overrunning clutch disconnects the gear train as soon as the engine fires, or when the test is completed. At this point, the 20 hp motor cuts-in to power a fan for the cooling cycle. The air-tube clutch controls this engagement. An air-on, spring-off brake stops the fan.

This is an installation of Trans World Airlines, Kansas City, Mo.

CHAIN DRIVES convert tractor to crop picker.



ENGINE-SPEED change alters speed of attached accessories accordingly.

Chain-driven conveyors eliminate field labor

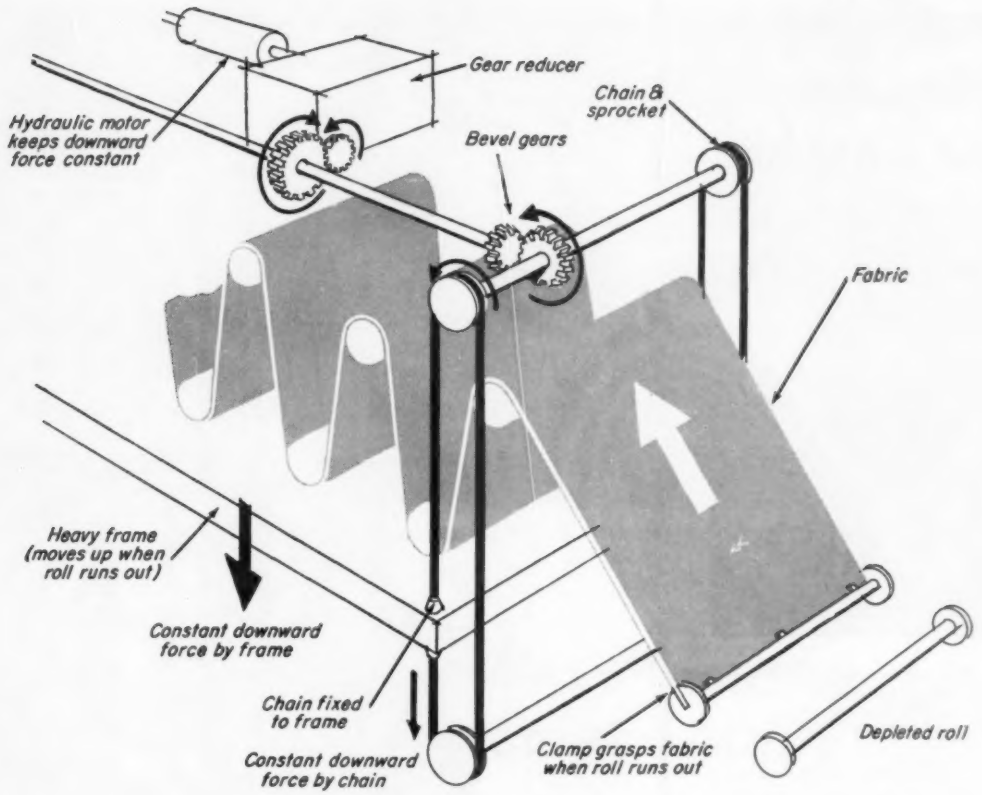
CHAIN DRIVES, MOUNTED to the power take-off of a farm tractor, eliminate considerable field labor and convert the machine to a one-man crop picker.

The chains drive a pickup rake, lateral conveyor, side-conveyor, and fans and blowers. The frame on which the sprockets, pillow blocks, shafts and gear box are fixed, is designed for seasonal mounting and dismounting to free the tractor for off-season work.

Natural synchronization of all operations is achieved. As the tractor engine-speed is changed, speed of all accessories changes too.

This gives the operator constant control over his equipment, allowing him to use all the accessories without re-adjustment. This saves considerable time and labor, cutting cost of any project using the tractor.

Designed by A. D. Goodwin and Sons, Inc., Manteca, Calif., the drives are shown on a Massey-Ferguson tractor for picking up almonds, walnuts, pecans and other crops.


Gear reducer
Chain & sprocket
Hydraulic motor keeps downward force constant
Bevel gears
Fabric
Heavy frame (moves up when roll runs out)
Constant downward force by frame
Chain fixed to frame
Constant downward force by chain
Clamp grasps fabric when roll runs out
Depleted roll

MOTOR TORQUE aids downward weight of frame. This keeps constant tension on fabric. When fabric roll runs out, frame moves up and motor goes out of stall, balancing forces on fabric.

Hydraulic motor keeps fabric tension constant

A HYDRAULIC MOTOR with 1200 in-lb torque, resists upward movement of a weighted bed, keeping tension of fabric constant during play-out. The motor and bed maintain a constant 3000 lb drag on the fabric being processed for the rubber industry.

When pull drops below 3000 lbs, the motor transmits power through a herringbone gear reducer to a pinion and gear. This turns the main shaft which is 4½ inches in diameter and equalizes the load.

When the fabric roll runs out, the reel-end is clamped and increases tension. This brings the motor out of stall, into reverse, and rebalances the load.

When a new roll of fabric is spliced and fed through the machine, the weighted bed again drops down, increasing motor's load, forcing it once more into stall.

Special feature of the machine: 12-inch diameter right-angle gears which transmit power from main to secondary shaft.

Standard sprocket turns 2½-inch pitch chain on which the bed is anchored.

Important development: connecting motor to check valves in the outgoing oil lines. This allows oil to be drawn in when motor is forced into reverse, maintains constant pressure.

Advantage: Using the motor as an assist when threading fabric through the machine, or doing maintenance work.

The device is a development of Giffels and Rossetti, Detroit, Mich.

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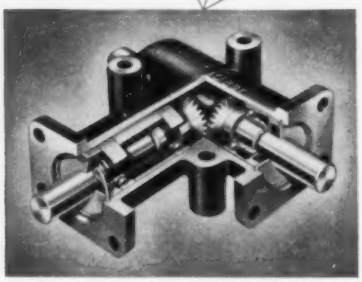
YOUR 90° DRIVE
PROBLEMS



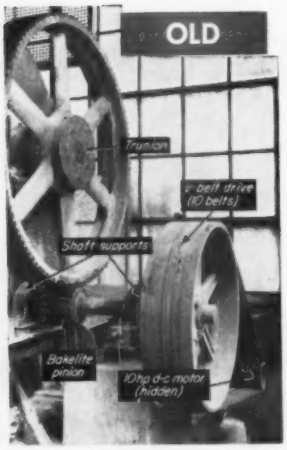
Hardened bevel gears and antifriction bearings insure smooth, trouble-free operation. Completely enclosed, permanently lubricated ANGLgear requires little or no maintenance.

Addition of the new 2¼ hp size to the ANGLgear line makes it easier than ever to solve your 90° power takeoff problems with this standardized right-angle drive. You can now specify ANGLgear in any one of 16 different stock models, offering power ratings from ⅓ to 5 hp, with choice of 1:1 or 2:1 gearing and 2 or 3-way shafting.

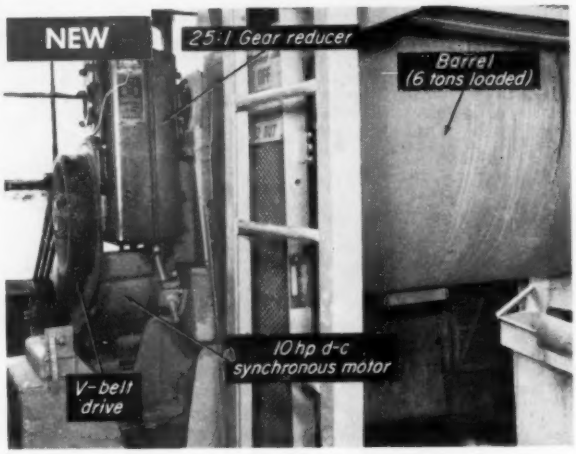
Compact, high capacity ANGLgear offers you several distinct advantages over V-belts or chains and sprockets. Featuring 5-way mounting, it is easier to design into your power transmission systems. Incorporating positive bevel gear drive, it eliminates slippage and backlash problems. Completely enclosed, it presents no safety hazards. Permanently lubricated, it requires virtually no maintenance. And not the least important, ANGLgear invariably costs less than other types of 90° drives.

See our literature in Sweet's Product Design File or contact your local distributor.

AIRBORNE®
AIRBORNE ACCESSORIES CORPORATION
HILLSIDE 5, NEW JERSEY
Circle No. 5 on Reader Service Card



SPEED REDUCER is powered by 10 hp, a-c, synchronous motor, turns six-ton grinding barrel at 21 rpm. The reducer-gears, fully enclosed, are shielded from destruction by abrasive dust. The old drive used 10 belts, a counter-shaft and a Bakelite dry pinion. Working life is indefinitely longer, used to be six months.



Enclosed gearing stops abrasive wear

A SPEED REDUCER installed on an abrasive grain grinder stops costly replacement of exposed gears. Three V-belts now do the work of 10 and power the gearbox more efficiently at 25:1 speed reduction.

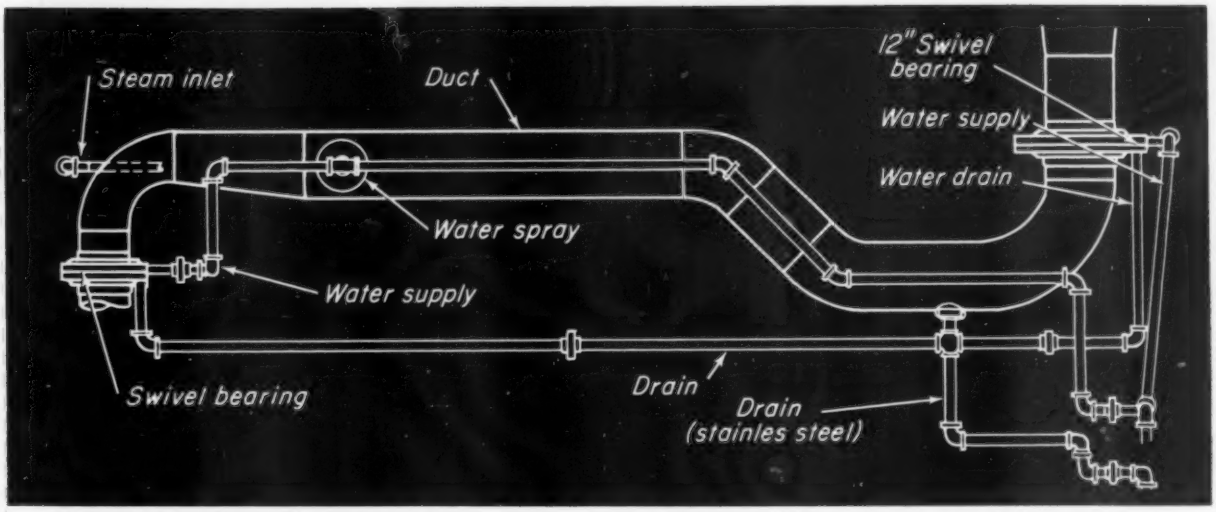
Vibration and gear-wear were once costly. It was impossible to lubricate the exposed gears. Grease and oil picked up fine particles of dust, mixed them into a destructively-abrasive paste which increased gear and pinion wear.

The process involves grinding abrasive grain in a barrel, 5 ft. long by 5 ft. dia. Loaded, it weighs 6 tons, and turns at 21 rpm. It is driven by a 10 hp a-c synchronous motor.

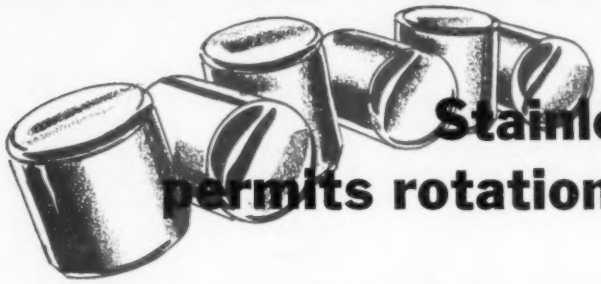
With 10 V-belts, sheave diameters were: 30 inches at 144 rpm and 6 inches at 720 rpm. The small gear had a pitch diameter of 7.2 inches at 144 rpm; the large gear had an OD of 48.5 inches at 21 rpm.

Now using three V-belts, sheave sizes have changed to: 12 inches dia. at 720 rpm for the small; 16 inches dia. at 540 rpm for the large. Speed reduction on belts is 1.335 : 1.

This installation was made at the Electro-Minerals Division of the Carborundum Co., Niagara Falls, N. Y.



STAINLESS BEARINGS make adjustment easy in any direction, reach every part of motor-test area for exhaust.



Stainless bearing permits rotation at 1400°F

HEAT AND ACID RESISTANT BEARINGS allow rotation at 1400°F temperatures in a mist of H_2SO_3.

Designed of stainless steel, in sets of 6", 9", and 12" diameters, the four-point contact bearings carry a maximum of 175,000 in-lb, allowing adjustment of an engine-test exhaust system serving 26 motors from 200 hp to 1500 hp —running at full throttle for 24 hours.

The stainless steel bearings, fitted on two swivels, allow conduits to reach engine-exhaust ducts wherever positioned in the test-area.

Heat inside the system rises from room temperature to a steady peak of 1400°F.

Goal was to develop a bearing giving perfect adjustment of the swinging exhaust-arm under these highly destructive conditions.

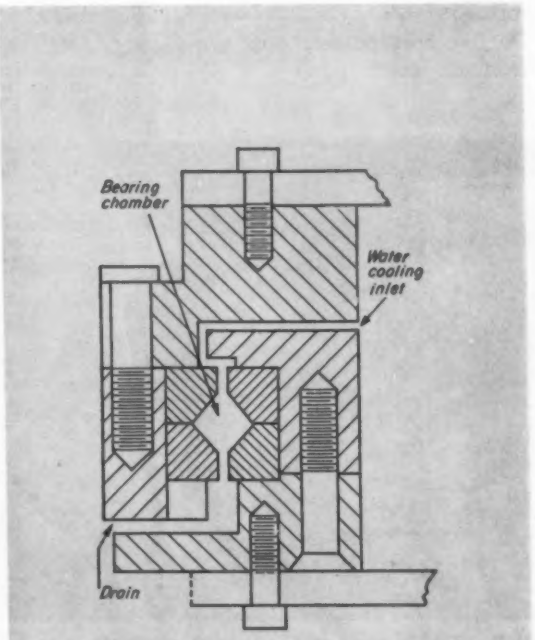
Problem 1: Pick a bearing that would resist heat.

Solution 1: Blow water-cooled air over set of bearings farthest from the engine. The other set could not be reached.

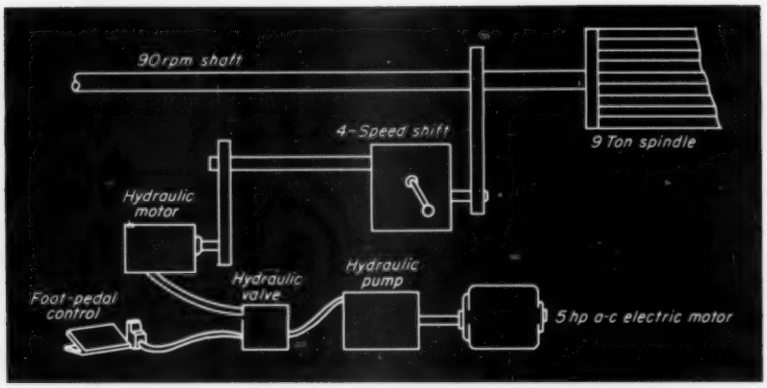
Problem 2: When moist air is blown over bearings, water and fumes combine, making sulphurous acid spray.

Solution 2: Using stainless steel bearings.

The bearings are the development of the Detroit firm of Giffels and Rossetti.



BEARINGS are housed in chambers and cooled with water.



VARIABLE CONTROL of wire-winding roll is made possible by using foot-pedal to control fluid flow to hydraulic motor

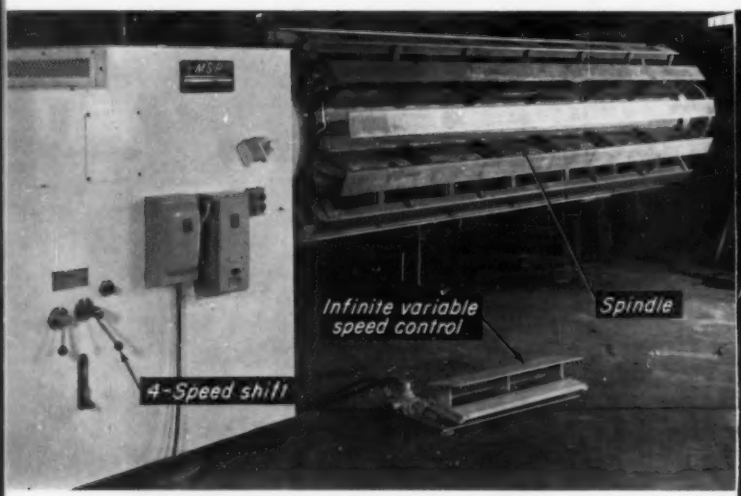
Hydraulic motor gives Infinitely variable speed control

BY REGULATING OIL FLOW between a hydraulic pump and hydraulic motor, infinite speed control of a coil winder is achieved.

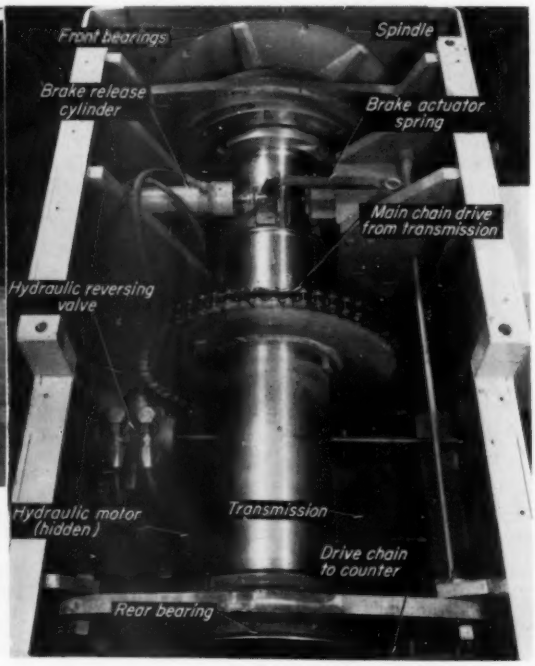
Power from a hydraulic motor is transmitted through a four-speed transmission to a 9-ton spindle which winds wire at 90 rpm.

A five-horsepower, a-c, synchronous motor powers a hydraulic pump which forces fluid to a hydraulic motor at constant speed. Between pump and motor is a hydraulic valve operated by the machine's operator with a foot pedal. It gives him constant control over fluid going to the motor.

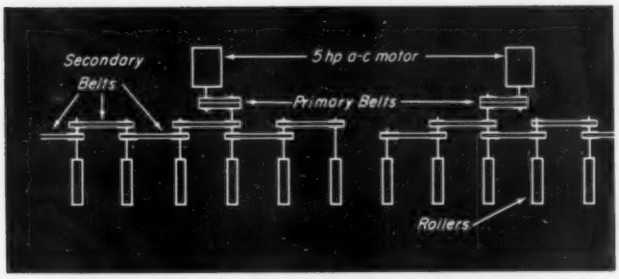
Coils which are wound on the machine's spindle must be of specific weights, exact lengths, and use a certain type of wire. Problem facing the manufacturer was: How to guide the wire, count the exact number of turns, trigger the shut-off mechanism, and achieve infinite variation with control. Using hydraulic power allowed this to be accomplished without converting to d-c operation. Machine is the development of Mason, Shaver, and Rhoades, Inc., East McKeesport, Pa.



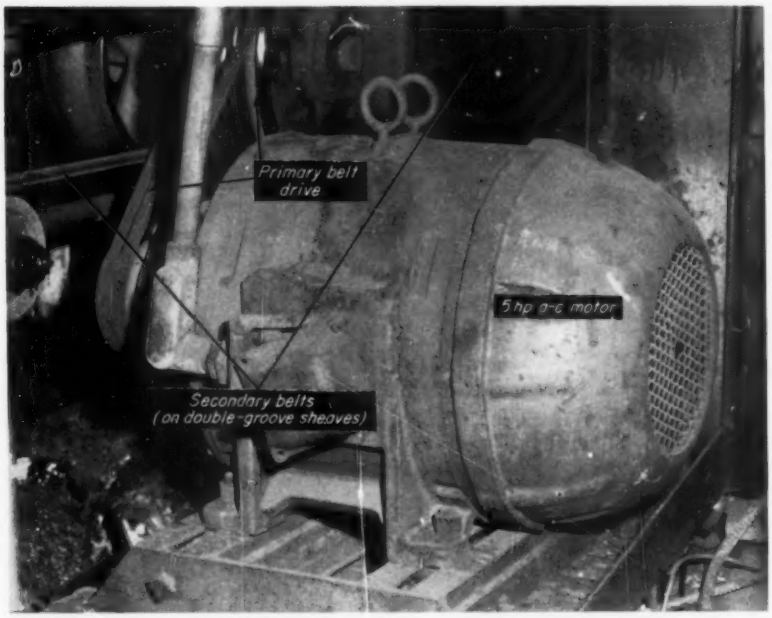
FOOT PEDAL gives the operator complete command of his machine's speed at all times.



POWER TRANSMISSION unit is seen uncovered showing bearings, brake and main chain drive.



POWERING RUN-OUT TABLE, two of these 5 hp, a-c motors do work once done by two 5 hp, and one 7 hp motors. Rubber belts replaced gears; weight is reduced and the 7 hp motor is eliminated. No gears means no lubrication and no gear replacement.



MAINTENANCE COSTS DROPPED 95% when pair of 5 hp, a-c motors were installed on table-rollers, powering them with V-Belts. Replaced were: 40 pairs of right-angle gears, three line-shafts, and three electric motors.

Belt drive cuts upkeep 95%

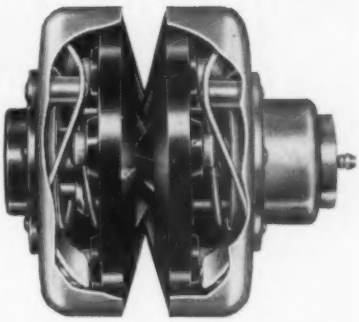
MAINTENANCE COSTS DROPPED 95% with the installation of 40 V-belts on a bar-mill runout table powered by two 5 hp, a-c, synchronous motors.

Replaced were: line shaft; two 5 hp motors; a 7 hp motor; and 40 pairs of bevel gears. Steel bars coming out of the furnace, move along the runout table and keep temperatures as high as 1200°F. Belts remain cool enough to operate efficiently and turn rolls placed five feet apart for a length of 200 feet.

Big problem with gears was maintenance. Each of the 80 gears was replaced yearly. Each pair was lubricated daily.

Now lubrication is done weekly. One V-belt has been replaced in a two-year period. Maintenance costs have been cut to a bare minimum. This is an installation of the Buffalo Steel Corp., Tonawanda, New York.

Hi-Lo Load-O-Matic Control Eliminates Pulley Slow Down



Hi-Lo Variable Speed Pulleys positively maintain the desired speed ratio over a wide range of load variation by means of an exclusive cam and cam follower assembly. This means:

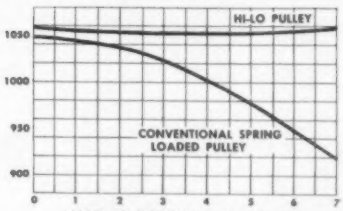
Pulley speed is independent of load and load changes. "Drag" is eliminated and high shock absorbency provided.

Pulleys do not compress belt due to spring pressure. Springs are not driving members. They act only to keep pulley faces in contact with belt.

Pulleys automatically regulate belt tension. Because of the cam assembly, belt is never under more tension than required by the load.

Double cams maintain constant belt alignment.

HI-LO PULLEYS COMPARED TO OTHER VARIABLE SPEED PULLEYS



PLUS THESE OTHER FEATURES:

- Smaller in size than comparable units.
- Quickly and easily installed.
- Replaceable face assemblies drastically cut repair and replacement costs.
- Available in sizes from .5 to 5 hp., ratios to 2.5/1 (single pulley) 6.25/1 (double pulley).

Request details and prices. Ask for Bulletin A-458.

Manufactured By HI-LO MANUFACTURING COMPANY

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NEW PRODUCTS

New belt drives save space

Space-savers are vital factors in machine design. The new line of Dodge V-belt drives seen at far right, features smaller, lighter sheaves and belts. Conventional drive is seen at left. Under trade name of Dyna-V, belts resulted from development of stronger sheave-metals, synthetic belt fibers and rubbers. Two sizes are made: 3/8 in for 1 hp to 50 hp; 5/8 in for 1 hp to 200 hp. A made-to-order series is for 1500 hp. Design includes crowned belt-top and concave walls.

Dodge Manufacturing Corp., Mishawaka, Ind.

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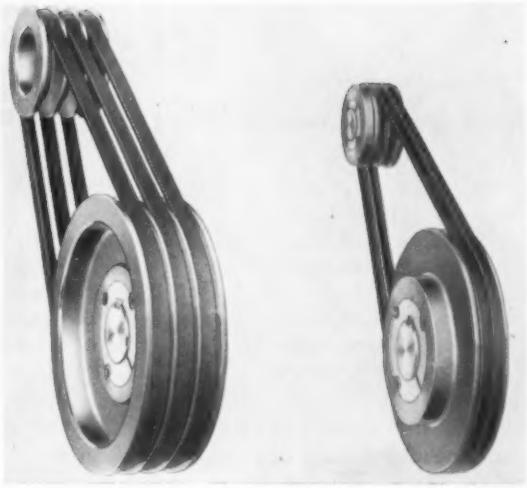


Split bearings erase need to remove shaft

Randall's new split bearing cuts bearing replacement costs by an estimated 75%. How? By doing away with the need for expensive shaft-removal operations. It is these which pyramid the costs of labor, transportation of parts, and worst of all—downtime!

This need means that whenever a shaft is removed men are standing idle, production stops, and maintenance costs string out. To solve these problems of cost and downtime, the split bearing was developed for intermediate position-installation and replacement. Because of the way it is designed, the bearing makes it possible to do this without re-positioning the unit or taking it apart. The Randall split-bearing is available in the following sizes: 1-3/16 in to 3-3/16 in. The two-piece, all bronze ball assembly is self-aligning.

Randall Graphite Bearings, Inc., Lima, Ohio.

circle number 225 on the reader service card

Mounting scoop saves time

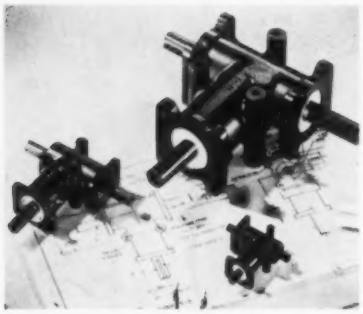
New motor-mounting assembly meets J.I.C. standards for foot-mounted, ball-bearing motors in industrial applications. The compact, scoop-mounted "Motodrive" is designed to support rigidly, a separately foot mounted, standard NEMA horizontal motor.

The heavy-duty cast-iron scoop-type platform is securely bolted to the drive frame, supporting the motor. Available sizes range from 1 hp to 10 hp. Others can be had up to 40 hp.

Reeves Pulley Co., Div., Reliance Electric and Engineering Co., Columbus, Indiana.

circle number 224 on the reader service card

More new products on next page



Right-Angle Bevel Gear Drive

Like the others in the 16-model line, the new 2¼-hp. ANGLgear comes with either 2- or 3-way transmission and with 1:1 or 2:1 shaft ratios. Maximum static torque rated 2,500 in.-lb. Speed rated 1,200 rpm. Measures 12 x 11¼ x 4 in. Mounts in nearly any position.

Airborne Accessories Corp., Hillside, N. J.

circle number 200 on reader service card

Variable-Speed Drive

A Servotran fitted with a solenoid-operated speed control will shift speeds in less than 0.1 sec. without disengaging either drive or load. With a new 2-solenoid unit, a Servotran will give 3 speeds and "off." Speed settings are accurate and repeatable within 0.5%. Efficiencies run 85% to 95%, horsepowers 1/70 to 1/4. Constant output torque.

Humphrey Products Div., Humphrey, Inc., San Diego, Cal.

circle number 203 on reader service card

Industrial V-Belt

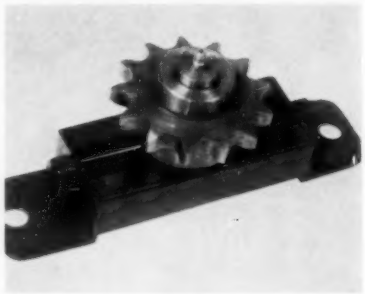
Cover: neoprene, to resist wear. Compression Member: fibers dispersed in neoprene. Makes belt flexible, yet transversely rigid, so it stays in place on the sheave. Reinforcing Cords: Dacron, to resist shock, wear, make belt plastic enough to simplify matching, save take-up time.

Boston Woven Hose and Rubber Co., Boston, Mass.

circle number 204 on reader service card

Roller-Chain Tightener

Two stock sizes. Idler sprocket, held in a welded steel frame,



is moved with an adjusting screw. You can make fine adjustments on the slack side of a chain even when it's running. Pressure lubricated.

Industrial Marketers, Ferndale, Mich.

circle number 206 on reader service card

Flexible Coupling

Six sizes handle 1/12 hp to 30 hp. You can get, for every size,

higher torque rubber sections. When overloaded, the rubber section slips, prevents damage.

Coupling Div., Guardian Products Corp., Michigan City, Indiana.

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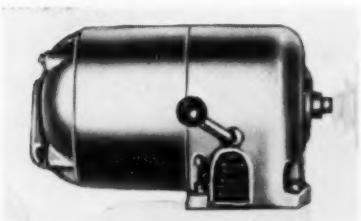
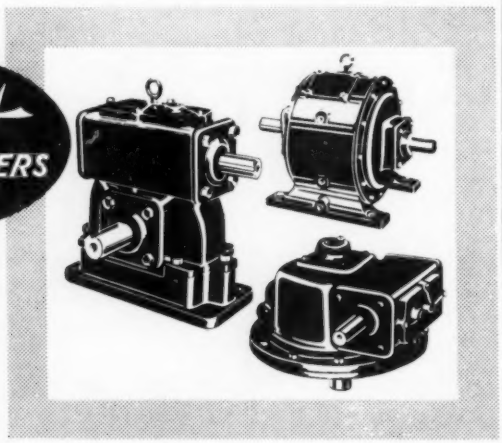


Vertically-Mounted Gearshift

Deliver full rated horsepower at all output speeds. For shaft-up mounting there are four types with integral motors and there is a motorless one to fasten to your own motor. The integral-motor types come in a variety of

Abart SPEED REDUCERS



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Chicago 50, Illinois

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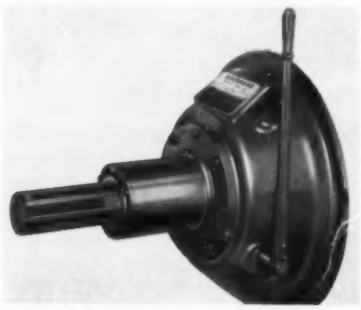
ratings and gear ratios. Fine for driving metalworking machinery, material handling equipment, tire machinery.

Lima Electric Motor Co., Inc., Lima, Ohio.

circle number 208 on reader service card

Power Takeoff

For extra-heavy-duty in oilfields or other industries, the troublemaking pilot bearing is eliminated along with bearing supports and flexible couplings. Reason is that the machine is very heavily built and main bearings are wisely placed. They're rated at 40,000 hours,



need lubricating only once a year. Clutch release bearing is lubricated for life. Drive shaft handles loads to 5,000 pounds. Comes with single- or double-plate, organic- or Morlife cera-metalic-faced, gear-tooth-drive clutches.

Rockford Clutch Division, Borg-Warner Corporation.

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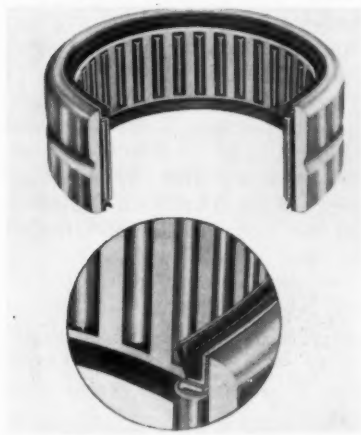
Sealed Cage-Type Needle Bearings

Neoprene seals hold in the lubricant, keep out the dirt. Come



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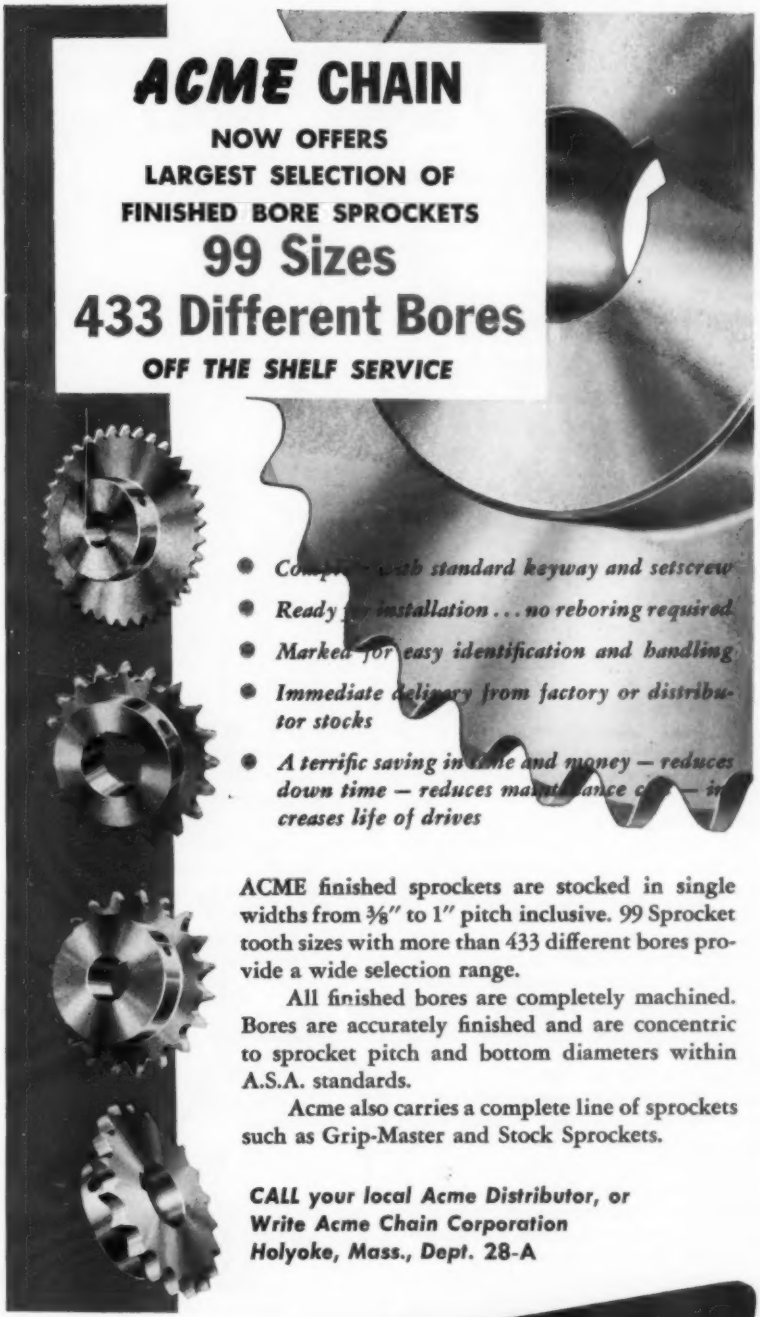

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Acme also carries a complete line of sprockets such as Grip-Master and Stock Sprockets.

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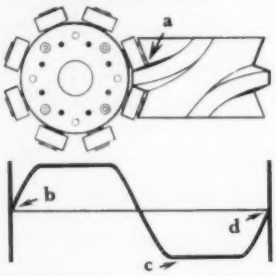
Critical Points in Acceleration Characteristics of Indexing Devices

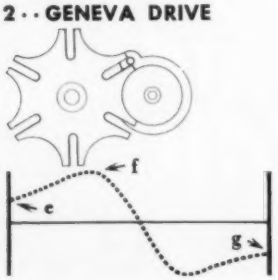
Maximum speeds of production machines are frequently affected by performance limitations of their indexing mechanisms. The efficiency of the indexing mechanism, itself, is dependent upon the amount of shock, vibration and other dynamic forces involved in the starting, stopping and rate of change from maximum acceleration to maximum deceleration.

FIG. 1 · · FERGUSON DRIVE



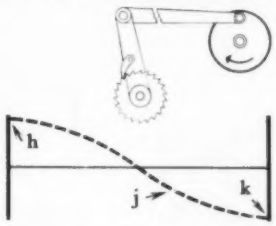
The acceleration curves of such mechanisms provide a graphic means of evaluating and comparing dynamic conditions that affect the performance of the indexing device and the machine. The Ferguson Drive in Fig. 1 is a cam indexing mechanism employing preloaded bearing followers rolling along a tapered cam rib. Two followers are in contact with the rib at all times (a) maintaining a zero backlash condition and an accuracy of .001" without auxiliary locating devices. This Drive features a modified trapezoid acceleration characteristic . . . a combined cycloidal and gravity curve. The Ferguson Drive has an initial force of zero (b) and a smooth, sinusoidal increase. Its maximum acceleration value (c) is very low and the change through maximum deceleration to zero (d) is gradual. These smooth indexing characteristics of the Ferguson Drive permit operation of production machinery at speeds as high as 2,000 RPM, if necessary, with a minimum of 8,000 hours operation while maintaining extreme precision and zero backlash.

FIG. 2 · · GENEVA DRIVE



The six-stop geneva drive shown in Fig. 2 is representative of the geneva family in that its dynamic forces are instantaneously applied and the initial acceleration force is fairly high (e) with an even higher maximum acceleration (f). The change to maximum deceleration is gradual but there is an almost instantaneous stop (g). When operated at high speeds, geneva drives wear out quickly and cause heavy vibrations that seriously affect the life of the machine and product quality.

FIG. 3 · · RATCHET & CRANK



The rachet and crank has a distorted cosine acceleration curve (a true cosine curve is illustrated in Fig. 3) with an extremely high initial acceleration (h). The descent to maximum deceleration (j) is gradual but, as with the geneva drive, stopping is instantaneous (k). Even with auxiliary braking action, overtravel is extreme and accuracy is very poor. Rachet and crank devices have very limited use in modern production machines.

FIG. 4 · ·

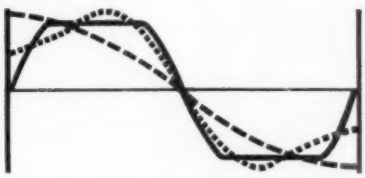


Fig. 4 shows the curves of the three mechanisms on the same diagram. Note that only the Ferguson Drive has an initial force of zero and the lowest maximum acceleration value combined with a gradual change to maximum deceleration. *This is why the Ferguson Drive does what no other indexing device can do!*

• SEND FOR CATALOG

Complete information about the Ferguson Drive is available in the form of an easy-to-use catalog containing load ratings and dimensions of many standard units, drawings and installation photos. There is a copy for every design engineer . . . for your copy write: **Ferguson Machine Corp., 7818 Maplewood Industrial Court, St. Louis 17, Missouri.**

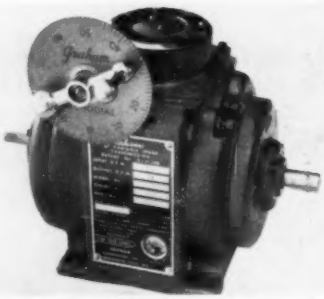
For more information, circle No. 38 on the Reader Service Card

NEW PRODUCTS *continued*

single or double sealed in two bearing lengths for ½- to 2-inch shafts. They added the seals without cutting roll length or load-carrying capacity and without changing the cage.

Orange Roller Bearing Co., Inc., Orange, N. J.

circle number 210 on reader service card



Micrometer-Dial Speed Control

Is precise, to 1 part in 4,000 on variable speed drive. Resets precisely and the setting holds because of a new anti-backlash control. Can be set at any speed, max to 0 in reverse, while running or stopped. Self-lubricated, overload protected, comes with or without built-in motor.

Graham Transmissions Inc., Menomonee Falls, Wisconsin.

circle number 211 on reader service card

Lubrication-Fitting Line

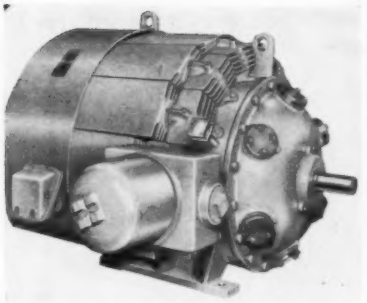
Relief-type fittings for bearing housings that have ¼-inch pipe thread openings. Used to come only in 1/8 inch. Tests show that these fittings save 80% of lubricating time. The cap revolves, and can't fall off. Can flip opened or closed with one finger. Body is flared inside.

Keystone Lubricating Co., Philadelphia 32, Penn.

circle number 212 on reader service card

D-C Heat Exchanger Motor

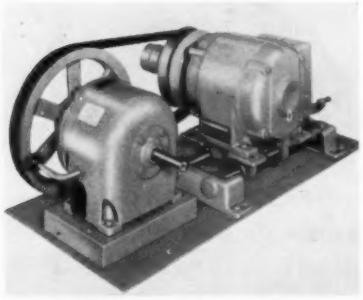
Mounts vertically, shaft up or down; mounts horizontally, base



up, down, or sideways. Explosion-proof or standard enclosures in 25 to 300 hp. ratings. Speeds adjusts from 5% of base speed through armature control (constant torque), to 400% of base speed through field control (constant hp.). A built-in a.c. fan, independent of the speed of the d.c. motor, cools constantly. It is responsible for much of the motor's small size and wide speed range. Heat exchange is air-to-air.

The Louis Allis Co., Milwaukee, Wisc.

circle number 214 on reader service card



Infinitely-Variable-Speed Drive

Comes in ratings of: ½, ¾, 1, 1½, 2, 3, 5, and 7½ hp.; speed variations of: from 12-to-1 as high as 25.7-to-1. It is a 1,800-rpm motor with a variable-pitch pulley.

Lima Electric Co., Inc., Lima, Ohio.

circle number 216 on reader service card

High-Strength Bakelite Insulator

Has two parts, and is sold together or separately. The base is of 16-gage, cadmium-plated steel. Insulator body is of bakelite, molded around a threaded brass insert. It is almost indestructible and meets military specifications. Designation: B1-1000.

Mullenbach Division, Electric Machinery Mfg. Co.

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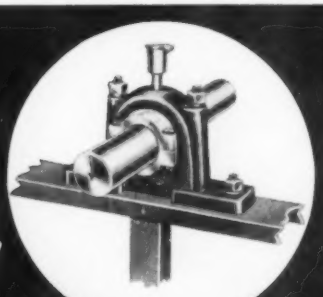
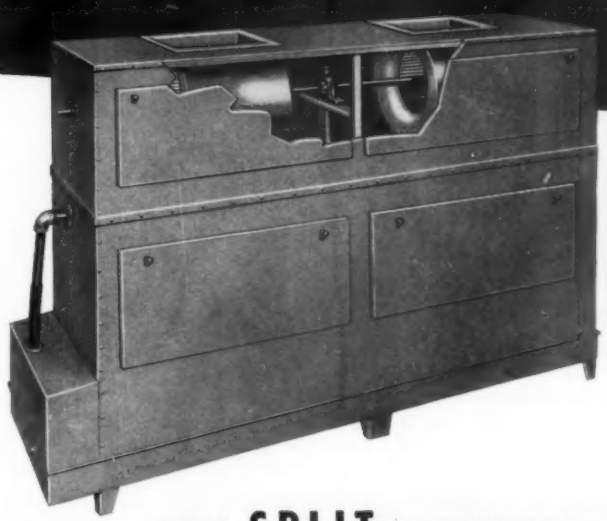
NEW

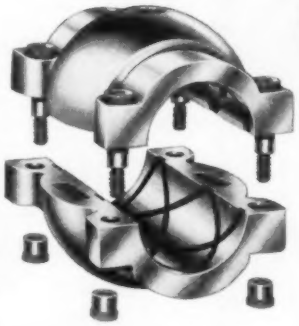
FAST-SIMPLE and EASY WAY to replace INTERMEDIATE BEARINGS





NEW SPLIT SLEEVE BEARING

CUTS REPLACEMENT COSTS *AT LEAST* 75%



ENDS SHAFT REMOVAL

This revolutionary new bearing introduces an unbelievable economy factor to industrial maintenance. Mounting is accomplished in a *fraction* of the time required for conventional solid bearings! The split feature permits mounting at any intermediate position on the shaft — *without* removing the shaft or disassembling the unit — and cuts installation time and costs because it allows for easy bearing replacement. This unit provides invaluable protection to manufacturers who offer warranty period replacement service.

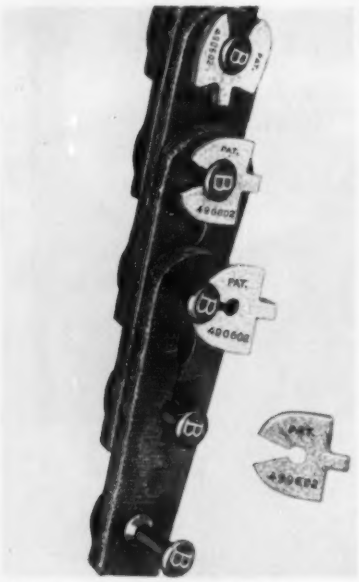
AVAILABLE IN POPULAR SHAFT SIZES FROM 1 3/16" TO 3 3/16" THROUGH AUTHORIZED DISTRIBUTORS— *(SEE YOUR YELLOW PAGES)*

Write for Supplement C of Catalog No. 57 for full details.

Randall

RANDALL GRAPHITE BEARINGS, INC.
LIMA, OHIO

For more information, circle No. 77 on the Reader Service Card



Washers For V-Link Belt

Made of Buffalo hide. Strengthens belt by protecting fabric of links and by stabilizing whip caused by pulsation. All sizes.

Brammer Corp., New York.

circle number 218 on reader service card

Ball Joints

Valuable for linkages. The old spring clip is replaced by a hardened, self-locking, hex-head setscrew. It locks the ball joint any-

place along a rod. Sizes to fit 1/4- and 5/16-inch rods.

Superior Ball Joint Corp., Fort Wayne, Indiana.

circle number 221 on reader service card

Self-Adjusting Disc Brakes

Rated at 125 to 575 lb-ft torque. Designed as "90M" series. No need to adjust for wear until the discs wear out. Brake's high thermal capacity makes it suitable for combination brake-motors. "Thru-shaft" construction, standard length Nema shaft extension.

Dings Brakes, Inc.

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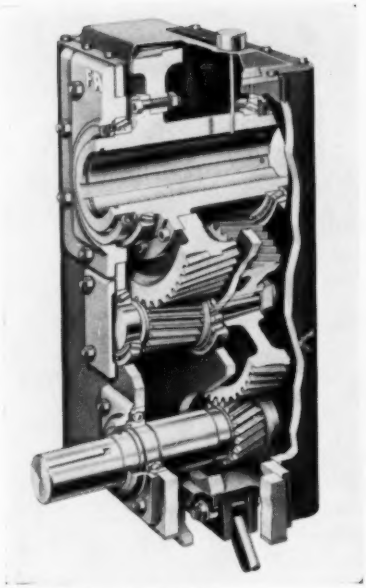


Shaft-Mounted Drive

All steel. Ratio 25:1. From 3 hp. at 5 rpm to 30 hp. at 50 rpm. Torque now 41,000 in.-lb. High-pressure-angle helical gears carry 12 to 15% greater load than ordinary helical gears. All steel 3-wall housing keeps moving

Gerbing

VARIABLE SPEED DRIVES

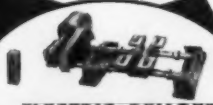
All V to V
Fractional To 25 HP
Ratios Up To 8 To 1

ROTO-CONE
- 1/8 to 25 HP
- Speed Ratios Up to 4 to 1

VAR'A'CONE
- For all 'A' and 'B' belt drives
- 1/8 thru 1 1/2 HP
- Ratios up to 2 3/4 to 1

ROTO-DRIVE
- 1/8 to 25 HP
- Speed ratios up to 8 to 1
- No adjustable motor base needed

VAR'A'CONE (Fixed Center Drive)
- For 'A' and 'B' belt drives
- 1/8 thru 1 1/2 HP
- Fixed shaft centers



ELECTRIC REMOTE CONTROL BASES
- Used where drive is out of reach of the operator

Complete selection of variable speed belts, adjustable and countershaft motor bases, controls, companion sheaves, and flexible couplings.

WRITE for 4 page Bulletin— AP-158.

Gerbing MANUFACTURING CORP.

11810 Milwaukee Ave., Northbrook, Ill. (Suburb of Chicago)

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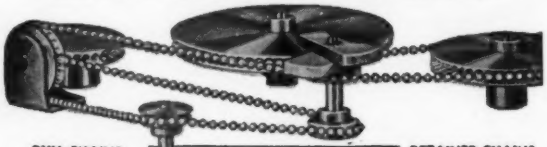
New! BEAD CHAIN® DRIVES

Low-Speed, Positive Drives or Motion Transfer ... *At Far Less Cost!*

NOW—BEAD CHAIN—QUALIFIED FOR DRIVES

Swivel-like Bead Chain and special non-slip sprockets make any drive possible ... at low cost ... eliminates expensive gears, universal joints, etc. Perfect for inaccessible locations!



PULL CHAINS

RETAINER CHAINS

Kinkless
BEAD CHAIN

Chain of a thousand uses! Low cost, attractive, easy to use! 5 Sizes: 3/32" to 3/8". Strengths 20 to 200 lbs.

Write for FREE Catalog!

SUSPENSION CHAINS

DOOR CHECK

TAG CHAINS

THE BEAD CHAIN MFG. CO.

209 Mountain Grove St., Bridgeport 5, Conn.

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parts aligned, withstands blows.

The Falk Corp., Milwaukee, Wisconsin.

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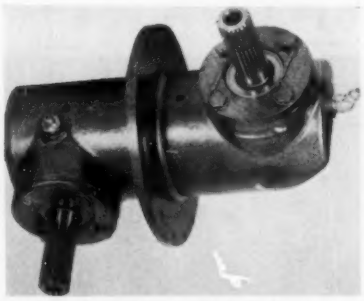

Right-Angle-Gear Control

For manual control of wing-flaps in aircraft. Meets full military environmental tests. Efficiency 95.5% — a 243-in.-lb. input gives a 232-in.-lb. output torque.

Precision Gear Division, Perfecting Service Co., Charlotte, North Carolina.

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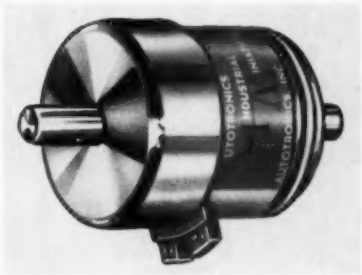

Electromagnetic Clutch

For tape recorders, calculators, coin changers. Standard voltages 24 to 28 d.c. Pilot mounting diameters: 1.125 in. Clutching torque: 40 in.-oz. No slip rings. Completely self contained. Both input and output shaft are free running but energizing couples them.

Autotronics, Inc., Florissant, Missouri.

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Miplon—New Teflon Blend

Has all Teflon's merits such as lubricity and inertness, but deforms 10 times less under load, wears 500 times better, has 4 to 5 times the compressive strength and stiffness.

Modern Industrial Plastics, Dayton, Ohio.

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Continued on next page

Smooth, quiet power transmission for heavy loads—

H & S Herringbone Speed Reducers

The Herringbone gear design provides continuous tooth action — eliminates end thrust.

Built for heavy load conditions, Horsburgh & Scott Herringbone Speed Reducers give you dependable economical service. They're available in single, double and triple reduction units. Check these 9 points of superiority:

1. Overall design conforms to AGMA specifications.
2. All bearing loads are balanced, due to the symmetrical design of the gearing.
3. Oversize bearings and low speed shaft provide tremendous overhung load capacity.
4. Heavy wall and base-pad thickness provides extra housing rigidity.
5. Housing designed with box-type construction for maximum thermal capacity.
6. Every gear is accurately sized and then cut on a modern Sykes continuous tooth gear generator.
7. All pinions are integral with the shafts and are made of heat treated alloy forgings.
8. Dust and oil-proof seals are provided on shafts extending outside the housing.
9. Splash lubrication floods all bearings and gears.

You'll find a wealth of information in our Catalog 55 describing our complete line of Speed Reducers. Write for it, or ask your nearby H & S representative.

THE HORSBURGH & SCOTT CO.

GEARS AND SPEED REDUCERS

5112 Hamilton Avenue
Cleveland 14, Ohio

For more information, circle No. 52 on the Reader Service Card

Circle No. 11 on the Reader Service Card

NEW
ROTO-GUARD*
SHAFT MOTION INDICATOR

Connects to rotating shaft

Actuates warning lights or other signals. Operates control switches.

PROVIDES positive indication of drop in speed or stopping of machinery

PROTECTS conveyors • elevators • feeders • belt conveyors

OPERATING SPEEDS of 10 to 150 rpm make the **ROTO-GUARD** applicable to automatic protection of slow-moving machinery used in material handling and similar systems

RUGGED AND COMPACT: approx. 5" x 6" x 6".

SIMPLE • POSITIVE • RELIABLE • LOW COST

A Development of
THE BIN-DICATOR COMPANY
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Specialists in Automatic Control Devices for Material Handling Systems for Over 20 Years
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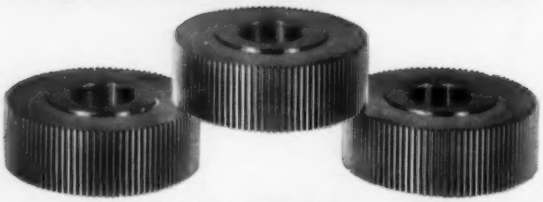
WE SELL DIRECT • PHONE ORDERS COLLECT

Ultra High Precision
Fine Pitch



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.0002 Maximum Composite Error
20 Pitch and Finer
14-1/2° and 20° Pressure Angles
Hardened and Ground Tool Steel
Quick Delivery on Non-stock Items

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2363 E. NINE MILE ROAD • HAZEL PARK, MICHIGAN
PHONE: SLOCUM 7-8840
Circle No. 59 on the Reader Service Card

Dry fluid drive absorbs peak belt loads

BY ADDING A DRY fluid drive to a sand and gravel conveyor unit, motor trouble, torn belts and considerable downtime are eliminated.

The belt is powered by a 30 hp, a-c, 1800 rpm motor. Sand and gravel from overhead hoppers are dropped onto the belt for mixing and conveying to gondola cars. Belt speed is 650 fpm to 700 fpm which is relatively fast for this kind of operation. A car can be loaded in approximately 3 min.

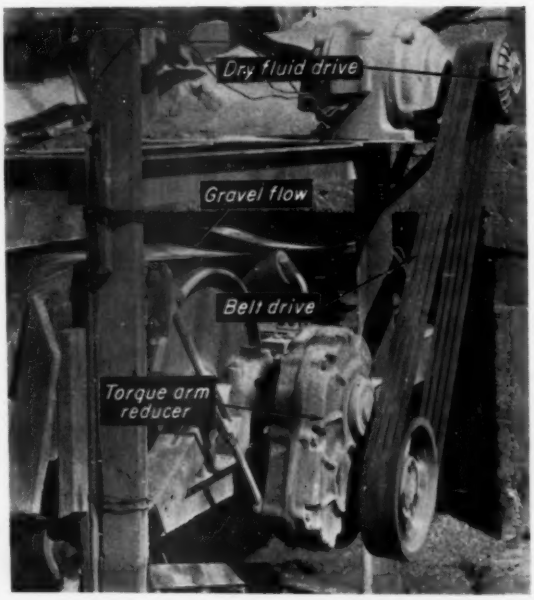
The drive lets the motor come up to speed instantaneously when starting; the conveyor belt comes up to speed slowly. No slip occurs at speed. The unit is protected from peak loads as high as 200%.

This is the result of accurate application of the use of a dry fluid drive.

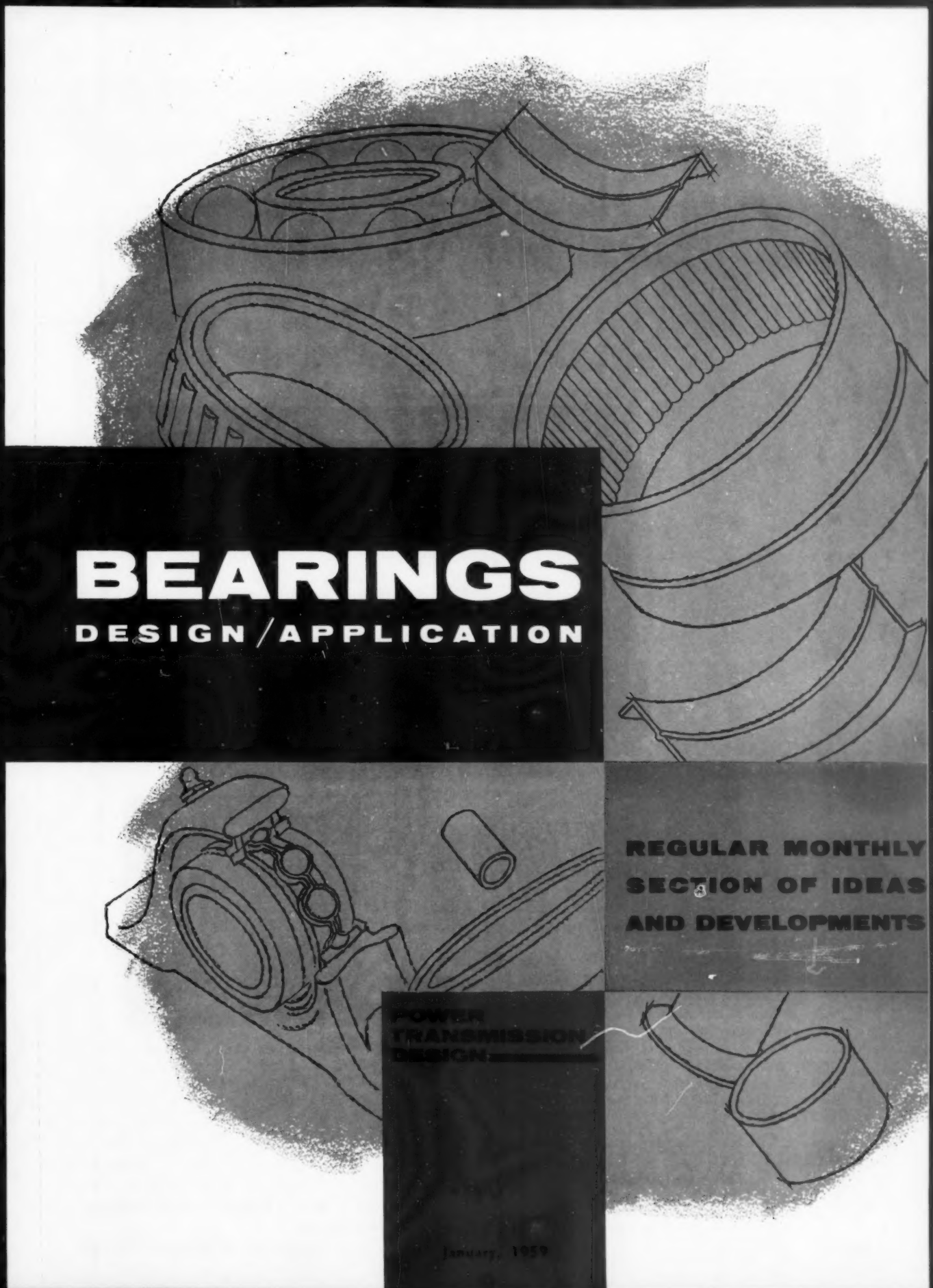
How it has overcome the difficulties met with in the sand and gravel conveyor, can be used as an example for applications in other industrial areas.

Outstanding is the fact that slip occurs only at starting and during excessive peak loads on the belt.

This is an installation of Wollcottville Sand and Gravel Co., Wollcottville, Ind.



DRY FLUID DRIVE stops motor trouble at gravel pit.



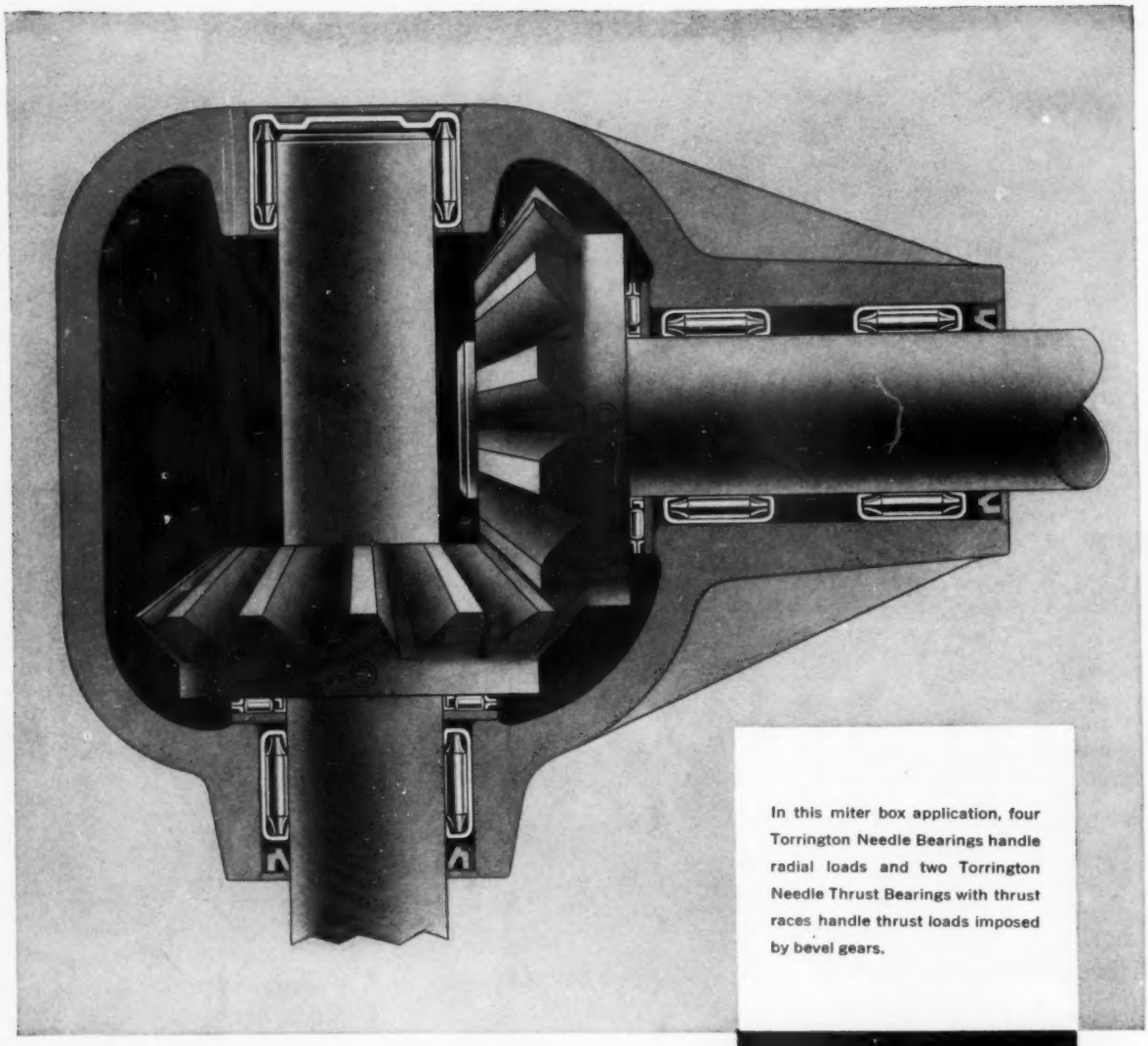
BEARINGS

DESIGN/APPLICATION

REGULAR MONTHLY SECTION OF IDEAS AND DEVELOPMENTS

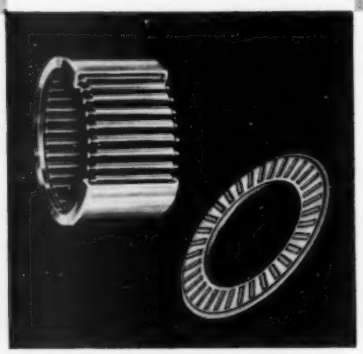
POWER TRANSMISSION DESIGN

January, 1959



In this miter box application, four Torrington Needle Bearings handle radial loads and two Torrington Needle Thrust Bearings with thrust races handle thrust loads imposed by bevel gears.

Perfect Combination for Thrust and Radial Loads



Here's a space-saving, cost-saving way to handle high thrust and radial loads. Just team up Torrington Needle Bearings with Torrington Needle Thrust Bearings!

With their full complement of small diameter rollers, Needle Bearings handle higher radial loads than any other anti-friction bearing of comparable cross section. And Needle Thrust Bearings are only .0781" thick – as thin as an ordinary thrust washer. Together they make a perfect combination of compact, light, rugged anti-friction bearings.

Either type of bearing may be run on hardened and ground adjacent parts to meet minimum space requirements. Or they may be used with standard races available from Torrington. To make the most of this efficient combination, call on our engineering staff for application advice.
The Torrington Company, Torrington, Conn.–and South Bend 21, Ind.

TORRINGTON BEARINGS

District Offices and Distributors in Principal Cities of United States and Canada

NEEDLE • SPHERICAL ROLLER • TAPERED ROLLER • CYLINDRICAL ROLLER • BALL • NEEDLE ROLLERS • THRUST

For more information, circle No. 83 on the Reader Service Card

Using anti-friction bearings?
Here are the pros and cons of . . .

Oil vs. grease

By **W. J. SCHIMBKE** *Metallurgist-Chemist, Aetna Ball and Roller Bearing Co*

ANTI-FRICTION BEARING lubrication requires knowledge of specific operating conditions. Factors other than quality and type of oil or grease must enter into selecting the methods to be used.

Expressed in general terms, the success of any bearing depends on proper lubrication. This results in:

(1) Reduction of friction between bearing races, rolling elements and cage or separator thus lowering rate of internal wear.

(2) Protection of bearing component-surfaces from rust and corrosion.

(3) Dissipation of operational heat.

(4) Protection of the bearing by excluding foreign matter.

No hard and fast rule can be laid down as to when oil or grease should be used, but care should be taken to see that any lubricant is supplied at the proper time and in the correct amount.

There are many applications in which either oil or grease could be used satisfactorily, but the prime consideration of either depends upon these conditions:

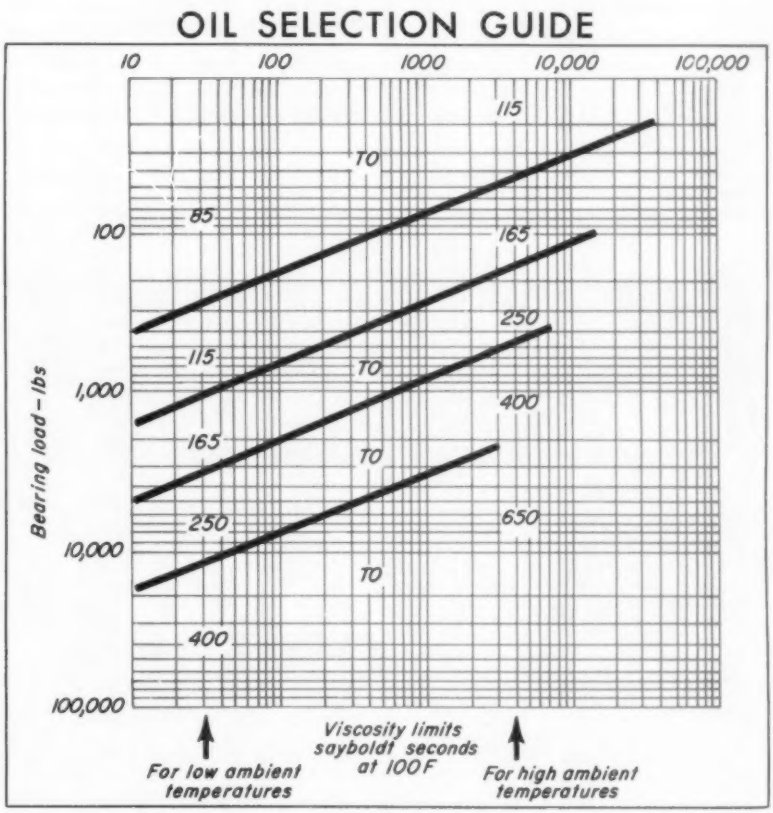
(1) Operational speed.
(2) Load carried.
(3) Accessibility for lubricating.
(4) Cleanliness in adjacent areas.
(5) Ambient temperature.

The most commonly recommended lubricant is oil and yet, in actual practise, grease is successfully used in a large majority of bearing applications. This is explained by the fact that there are definite advantages and disadvantages in using either type. By correlating the conditions with the advantages and disadvantages, proper selection is easy.

Oil Advantages

(1) Oil produces a flushing action on bearing surfaces—helps to wash off dirt or abrasive matter picked up during operation.
(2) Oil simplifies removal of lubricant and cleaning.

Continued on next page



(3) Oil develops less fluid friction internally than grease.

(4) Oil provides inherently cooler operation.

Oil Disadvantages

(1) Oil has tendency to leak through seals and contaminate surrounding area.

(2) Oil requires more elaborate sealing.

Grease Advantages

(1) Greater tenacity of grease enables it to stay in place without elaborate sealing.

(2) Grease provides sufficient lubrication for extended periods without renewal.

(3) With less frequent addition, grease reduces contamination by affording fewer opportunities for foreign matter to enter bearings.

(4) Grease permits use of simple seals which are often as effective as oil-lubricated bearings having elaborate seals.

(5) Grease provides a better protective film during periods of bearing idleness.

Grease Disadvantages

(1) Grease does not readily dissipate heat.

(2) Deteriorated grease is not readily removable.

(3) Grease is limited to low and moderate speeds (up to 3000 rpm).

Oil Lubrication

The most important characteristics of any oil are freedom from dirt or abrasive ingredients, resistance to chemical breakdown or oxidation and freedom from gum or varnish forming constituents.

Stable oils of mineral type are much preferred and are available in a wide range of viscosities to cover any type of application. Aircraft and automotive-type oils should be used because their better quality will, in the long run, prove relatively inexpensive in comparison with lesser quality oils which so often account for faulty lubrication and the consequent need of early bearing replacement.

Invariably oil is used when operating speeds are above 3000 rpm, when temperatures are above 200 F or below −40 F and when bearing housings are comparatively large and must be cleaned without stopping or disassembly.

The various methods of applying oil lubricants to bearings are:

(1) Drip feed. This method feeds oil to the bearings in regulated drops—assures constant lubrication.

(2) Felt or wickfeed. This method feeds oil by capillary action through a wick placed adjacent to the bearing, normally in a vertical position.

(3) Bath-type feed. This method requires an oil reservoir to be installed in the housing. Oil level should not be higher than the center of the lowest rolling element.

(4) Oil mist spray. Here, dried and filtered compressed air is used to force an oil mist through the bearing housing—a lubrication method particularly suited to cooler operation of extremely high speed applications.

There are various methods for applying grease, one of which is with grease cups. The most common type is a compression cup incorporating either a screw-down, spring-loaded or automatic device.

The screw-down type is used where hand-greasing is objectionable. It is however, an inefficient and wasteful method and is very easily contaminated during filling. Lubricant is supplied only when the cup is turned down by hand.

The spring compression-type is more efficient, since it keeps the lubricant under pressure, but pressure varies with the amount of grease in the cup.

The automatic type has a reservoir which can be filled by a grease gun. A spring-loaded plunger forces grease through a regulating valve plug and orifice.

GREASE SELECTION GUIDE

CHARACTERISTICS	METHOD OF TEST	OPERATING TEMPERATURE RANGE			
		0° TO +250°F	−65° TO +125°F	−40° TO +250°F	0° TO +300°F
Drop Point	A.S.T.M. D-566-42	300°F Min.	300°F Min.	300°F Min.	350°F Min.
Worked Penetration	A.S.T.M. D-217-52T	200 to 300	200 to 300	260 to 350	265 to 340
Oxidation Stability	A.S.T.M. D-942-50	Max. of 10 p.s.i. Drop in 100 hrs @ +210°F	Max. of 5 p.s.i. Drop in 100 hrs and 20 p.s.i. drop in 400 hrs @ +210°F	Max. of 5 p.s.i. Drop in 100 hrs @ +210°F	Max. of 5 p.s.i. Drop in 50 hrs @ +300°F
Evaporation	A.S.T.M. D-972-56	Max. of 3% loss in 22 hrs @ +210°F	Max. of 12% loss in 22 hrs @ +210°F	Max. of 2% loss in 22 hrs @ +210°F	Max. of 2% loss in 50 hrs @ +300°F
Oil Separation	Spec. Mil-G-3278		Max. of 4% in 50 hrs @ +160°F		Max. of 12% in 100 hrs @ +300°F
Low Temperature Torque	Spec. Mil-G-3278			Max. of 20,000 GM-CM-Sec @ −40°F	Max. of 30,000 GM-CM-Sec @ 0°F
High Temperature Performance	Spec. Mil-G-3278	Lubricate #306 bearing 4,000 hrs @ +212°F		Lubricate #204 bearing 1,000 hrs @ +250°F	Lubricate #310 bearing 2,000 hrs @ +300°F
Worked Stability 100,000 Strokes	A.S.T.M. D-217-52T			Max. penetration 375. Max. increase 100 pts	Max. penetration 375. Max. increase 100 pts
Water Resistance	A.S.T.M. D-1264-53T			Max. washout 50%	Max. washout 20%

Some types maintain an even flow as the spring tension decreases by decreasing the resistance to grease flow through the regulating valve.

Pressure gun greasing is the most prevalent and most convenient method of applying grease. Due to the danger of seal blow-outs, it is recommended that pressure relief devices also be used. Pressure relief-type fittings, which are commonly used, are equipped with a pressure-regulated orifice on the side of the fitting itself.

This feature prevents excessive pressure buildup during or after greasing.

Another type is a vent plug in the housing which is removed when grease is applied during operation. Excess grease exudes from the vent and, after a sufficient period of time, the proper amount of lubricant remains in the bearing.

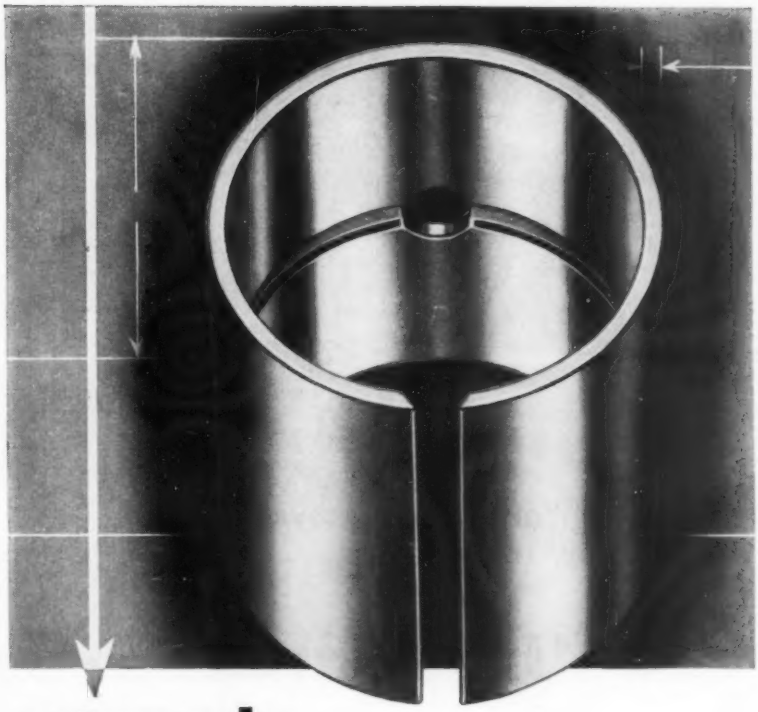
Where machinery is idle or cannot be operated while being lubricated, a simple control method involves recording the approximate grease capacity of the bearing against volume of grease discharged by each stroke of the gun.

There are applications where bearing housings are packed with grease. In these instances, the void area in the housings should not be packed more than 1/3 full for higher speeds and 1/2 full for slow speeds. It is also advisable to provide as large a grease space as possible, in order to keep an adequate supply of lubricant available.

A good rule is—a small supply of grease if continuously present is sufficient.

Excess grease causes churning within the bearing, temperature rise and volume increase. This can cause leakage and more rapid deterioration of the lubricant.

The greatest danger from using excess oil is churning and the subsequent generation of hydraulic pressure within the bearing. If this reaches a high enough value, seals are apt to leak or blow out.



extremely rigid specifications!

The illustration shows the cast bronze spindle bearing of a very well-known and popular metal-working lathe.

The manufacturer of this lathe has extremely rigid specifications covering tolerances, finish, structural quality of the bronze and roundness of the bearing even though the bearing is split for its entire length.

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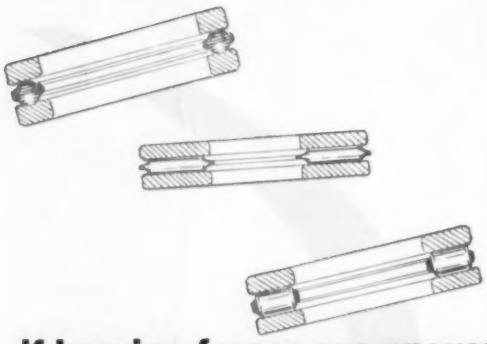
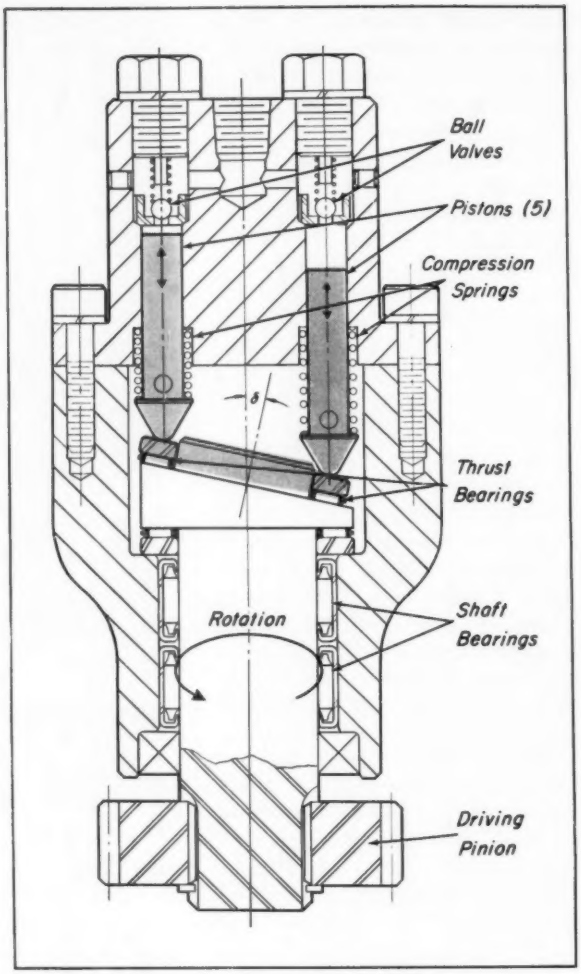


if bearing forces are uneven, here's how to calculate . . .

Unbalanced thrust loads

By **NORMAN M. WICKSTRAND**
Mathematician, The Torrington Co.



AXIAL PISTON PUMP with off-center and non-parallel load pattern.

Here are three ways to calculate effective loads on any type of thrust bearings. An axial piston pump is used as an example in which the load is applied off-center and non-parallel with the bearing axle.

To calculate effective thrust loads on an axial piston pump, five assumptions are made:

(1) Pistons are subject to full discharge pressure and the rest to zero gage. Since most pumps have odd number of pistons, the term one-half must be defined as whole number of pistons on one side of an arbitrary reference diameter, which produces worst load condition.

(2) Total load on bearing equals sum of load components of individual pistons acting as single load placed at centroid of loaded pistons.

(3) Bearing races are completely rigid except for local elastic deflection at areas contacting rolling elements. Deflection of each plus local deflection of contacting races is proportional to unit load applied.

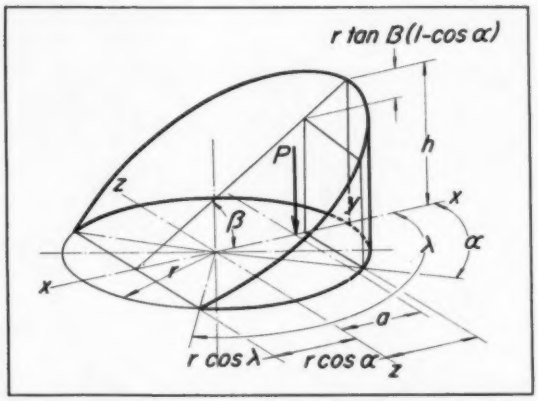
(4) When the bearing is needle thrust with large number of rolling elements, load distribution can be represented as smooth curve. Load intensity varies directly as normal distance from arbitrary reference chord.

(5) Limiting criterion for bearing service load is represented by maximum height of curved area. This is measure of heaviest load on any one roller or ball.

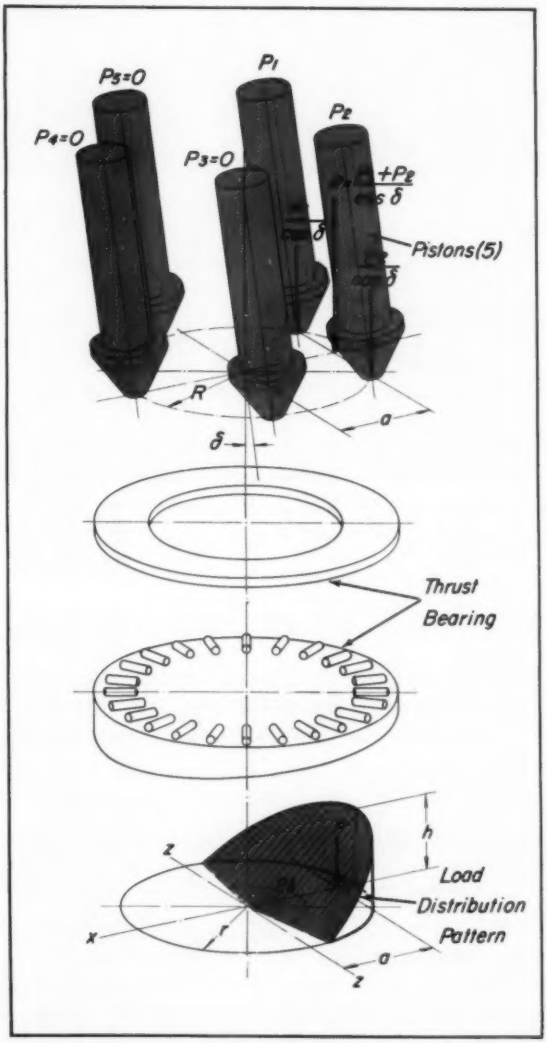
The section drawing shows that the centerlines of the main pump body and upper thrust bearing are inclined at a small angle δ to each other. The drawing on the opposite page shows an exploded view of the pump pistons and upper thrust bearing with thrust force components indicated for a five-piston pump.

From assumption 1 only, two pistons are loaded. The individual piston loads are labeled P_1 and P_2. These loads are coaxial. The component of each piston load, which is normal to the plane of the upper bearing, is $P_1/\cos\delta$ and $P_2/\cos\delta$. The angle δ is the same angle as the angle of the wedge or ramp. The angle is in the direction of the **zz** axis.

When the load is distributed as in assumptions 3 and 4, it can be represented by the shaded area shown. The load intensity is proportional



LOAD ON ROLLING ELEMENTS is distributed around portion of pitch circle.



to deformation, which varies directly as distance (measured along the **xx** axis) from reference chord **oo**. The shaded area can be described as the lateral area of an ungula.

The pump geometry shows that the pistons are placed along elements of a cylinder and that the lower piston tips are in a plane that is not normal to the axis of the reference cylinder.

Therefore, the trace of these tips is an ellipse. Since distances in the **xx** direction are useful, no error is introduced in assuming this trace to be a circle. This is particularly true since distances transverse to the ramp angle are used, and not up and down the ramp.

With an even number of pistons, the number of pistons under load may be either N/2 or N/2 − 1. With an odd number of pistons, the number under load may be either ½ (N + 1) or ½ (N − 1). For any given number of pistons, it is necessary to evaluate both conditions to learn which is the more severe.

The critical condition always will be where the positions are located symmetrically about the **xx** axis.

The total load on the bearing may be obtained from one of the following equations. The equation for the critical load is selected by trying both the first two or the last two.

When the total number of pistons is even

$$P = \frac{\pi d^2}{4} \quad \frac{p}{\cos \delta} \quad \frac{N}{2} \qquad (1)$$

$$P = \frac{\pi d^2}{4} \quad \frac{p}{\cos \delta} \quad \frac{N - 2}{2} \qquad (2)$$

When the total number of pistons is odd

$$P = \frac{\pi d^2}{4} \quad \frac{p}{\cos \delta} \quad \frac{N + 1}{2} \qquad (3)$$

$$P = \frac{\pi d^2}{4} \quad \frac{p}{\cos \delta} \quad \frac{N - 1}{2} \qquad (4)$$

Equation 2 represents the condition where two pistons are on the reference axis **zz**, but unloaded. This condition has less actual bearing load than if two pistons were considered loaded, but produces a more critical loading condition on the bearing due to load eccentricity.

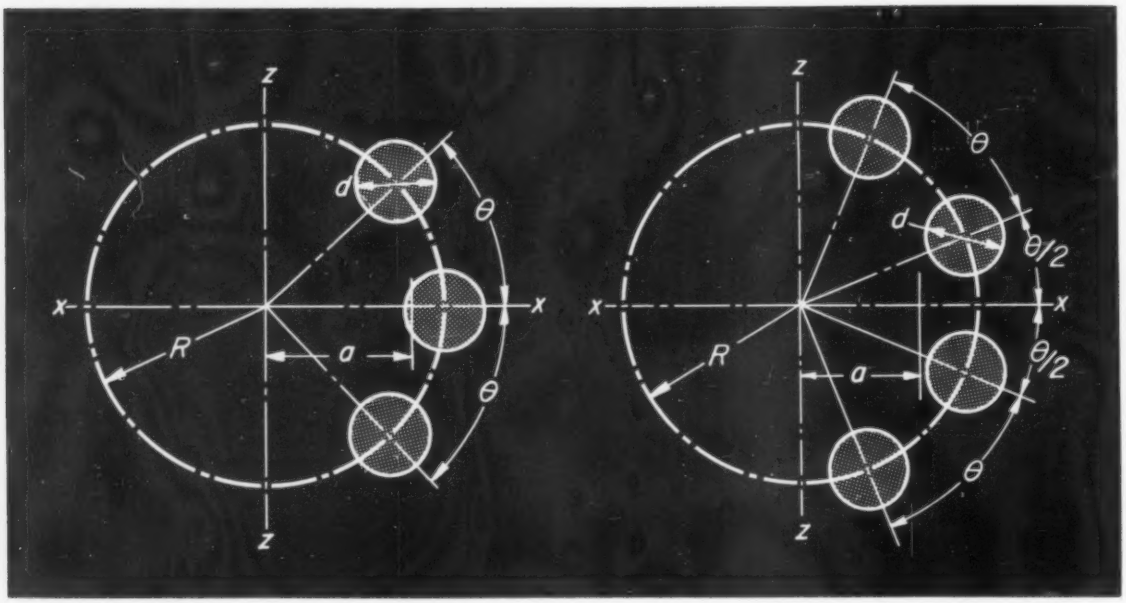
The line of action of the resultant piston load, P, is not co-axial with the thrust bearing or the pump, but is located at a point which is the centroid of the loaded piston-centers. The distance from the axis, **zz**, of the bearing to this centroid is indicated by dimension a.

To compute a for any condition, moments about the axis are equated. For each combination of total number of pistons and number loaded, a different equation must be set up and evaluated. For example, for a 9-piston pump, the equation is

$$P\,a = \frac{\pi d^2}{4} \quad \frac{p}{\cos \delta} \quad 2R \left(\cos \frac{\theta}{2} + \cos \frac{3\theta}{2} \right) \qquad (5)$$

Inserting the value of P from equation 4 and remembering that $\theta = 360^\circ/9 = 40^\circ$, the equation

(*Continued on next page*)



EVEN number of pistons . . . and odd number

reduces to a/R = .7199
Similarly, all values for a/R as given in table 1 are determined.

The load transmitted to the rolling elements of the bearing is distributed around a portion of the pitch circle as stated in the fourth assumption. The load intensity is assumed to vary directly as the normal distance from the reference chord. The curved surface of an ungula of a circular cylinder represents the load distribution. The maximum height h of the ungula represents the load on the heaviest loaded roller or ball of the bearing, and hence is used as the measure of effective bearing load.

The load transmitted to the rollers, as represented by the ungula, must equal load P applied to the bearing by the pistons. The centroids of the ungula and the loaded pistons must coincide at a distance a if the sum of moments about **zz** axis is to be equal. In other words, both the moment of the ungula and the moment of piston force must equal P a.

$$y = h - r \tan \beta (1 - \cos \alpha)$$

and

$$h = r \tan \beta (1 - \cos \lambda) \qquad (6)$$

The negative sign is used for $\cos \lambda$ because when λ is greater than 90° the cosine is negative. The above two equations combine to

$$y = r \tan \beta (\cos \alpha - \cos \lambda) \qquad (7)$$

Since P represents the curved area of the ungula

$$P = 2 \int_0^{\lambda} y\, r\, d\alpha$$
$$= 2r^2 \tan \beta \int_0^{\lambda} (\cos \alpha - \cos\lambda)\, d\alpha$$
$$= 2 r^2 \tan \beta (\sin \lambda - \cos \lambda) \qquad (8)$$

Moments about the **zz** axis are equated

$$P\ a = 2 \int_0^{\lambda} y r^2 \cos\alpha d\alpha = 2r^3 \tan\beta \int_0^{\lambda} (\cos^2\alpha - \cos\alpha \cos\lambda) d\alpha$$
$$= r^3 \tan\beta (\lambda - \sin \lambda \cos\lambda) \qquad (9)$$

By solving equation (8) for tan β

$$h = \frac{P\ (\sec \lambda - 1)}{2\, r\, (\tan \lambda - \lambda)} \qquad (10)$$

Assuming that the effective load. P_e, is equivalent to a uniform load of intensity "h" on the entire pitch circle

$$P_e = 2\pi r h$$

Inserting the value of h from equation (10)

$$P_e = \frac{\pi P (\sec \lambda - 1)}{\tan \lambda - \lambda}$$

The effective load is proportional to the applied load

$$P_e = F_1 P \qquad (11)$$

where F_1 is a correction factor.

Therefore, the value for F_1 in terms of angle is

$$F_1 = \frac{\pi (\sec \lambda - 1)}{\tan \lambda - \lambda} \qquad (12)$$

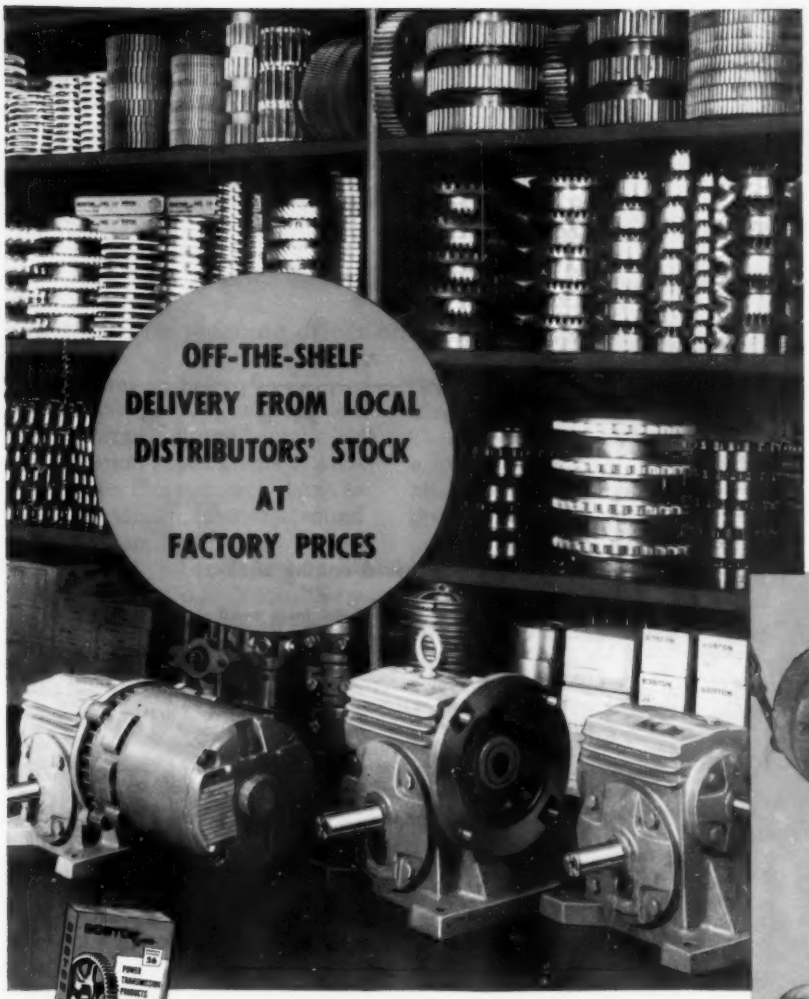
Equations (8) and (9) may be combined

$$\frac{a}{r} = \frac{\lambda - \sin \lambda \cos \lambda}{2 (\sin \lambda - \lambda \cos \lambda)} \qquad (13)$$

Equations (12) and (13) cannot be combined to eliminate λ but can be solved numerically.

Table 2 gives representative values of solutions which cover complete range of probable values. For any given value of a/r, the corresponding value of F_1 may be found. Values of angle λ are also given. Other values may be found by interpolation.

(*Continued on page 78*)



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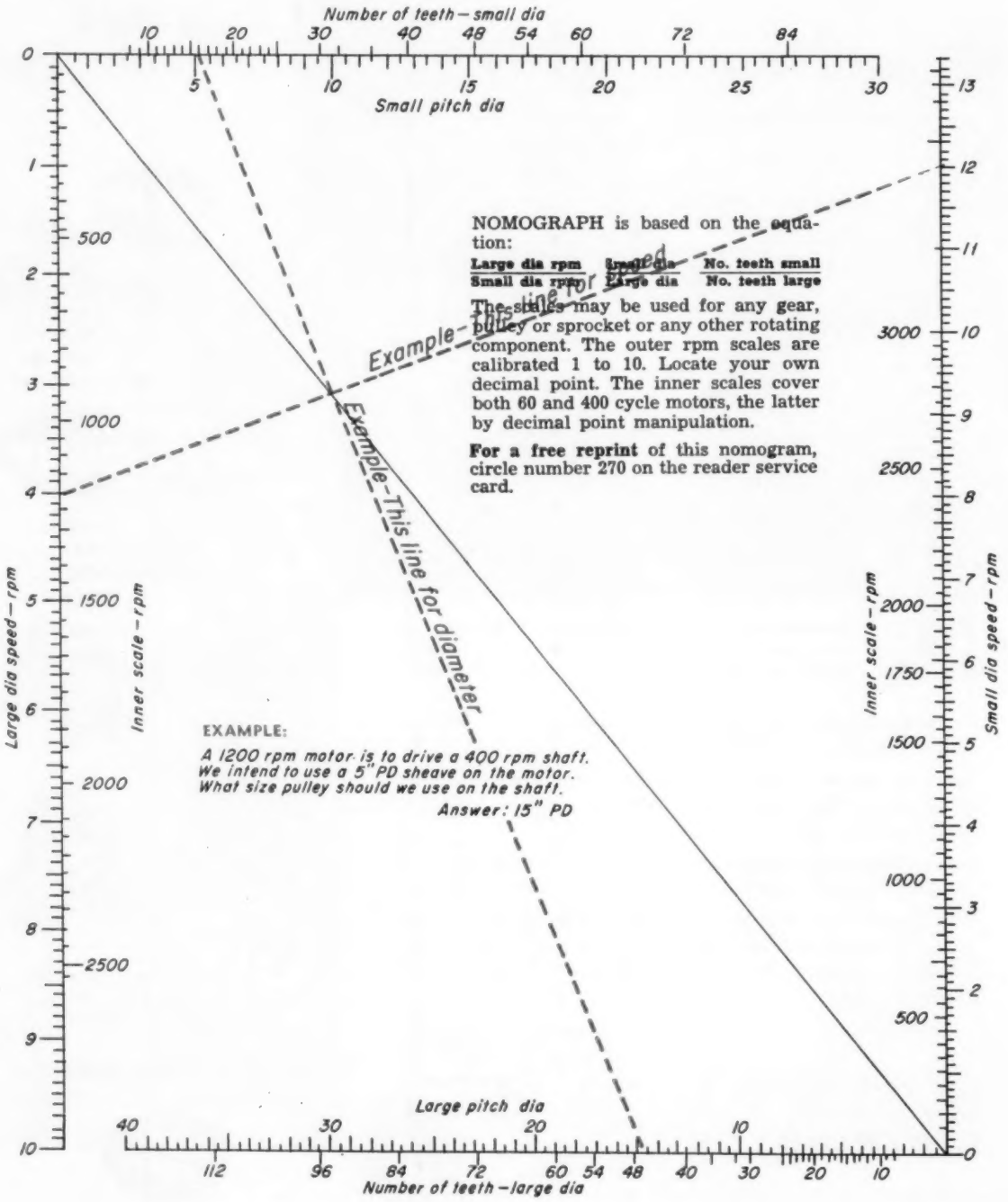
REFERENCE FILE

Pulley or gear speed vs. diameter

By **GEORGE B. RICHTER** *University of Minnesota*


Number of teeth – small dia
10 20 30 40 48 54 60 72 84
Small pitch dia
5 10 15 20 25 30
Large dia speed – rpm
Inner scale – rpm
Small dia speed – rpm
Large pitch dia
Number of teeth – large dia
112 96 84 72 60 54 48 40 30 20 10
Example – This line for speed
Example – This line for diameter
EXAMPLE:
A 1200 rpm motor is to drive a 400 rpm shaft.
We intend to use a 5" PD sheave on the motor.
What size pulley should we use on the shaft.
Answer: 15" PD

NOMOGRAPH is based on the equation:

$$\frac{\text{Large dia rpm}}{\text{Small dia rpm}} = \frac{\text{Small dia}}{\text{Large dia}} = \frac{\text{No. teeth small}}{\text{No. teeth large}}$$

The scales may be used for any gear, pulley or sprocket or any other rotating component. The outer rpm scales are calibrated 1 to 10. Locate your own decimal point. The inner scales cover both 60 and 400 cycle motors, the latter by decimal point manipulation.

For a free reprint of this nomogram, circle number 270 on the reader service card.

LITERATURE

on machine drives and related components

Endless woven belts
Descriptive catalog sheets are now offered, giving details of transmission applications of endless woven belts on such items as metalworking machinery, sound reproduction equipment, computers, and others. Arthur S. Brown Mfg. Co., Tilton, New
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Integral horsepower motors
Bulletin 104, eight pages, gives details on open and closed, polyphase and single phase motors adapted to special applications. Components, including bearings, end-frames, etc. are described. The Leland Electric Co., Dayton, Ohio.
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Multiple V-belts
An excellent 32 page booklet, number S-5132, titled "Goodyear Engineering Data For Multiple V-Belts", containing such needed data as "Fundamentals of V-Belt Design", and tables on "V-Belt Size for Required Horse-Power Rating". Goodyear Industrial Products, Akron, Ohio.
circle number 107 on reader service card.

Coupling
Solving mis-alignment problems is explained in this four-page bulletin titled "Dykman Sphere Gear Shaft Couplings". Sheets inserted give list prices, dimensions and ratings, capacities, and tips on coupling-selection.
The Dykman Mfg. Co., New York, N. Y.
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Clutches and brakes
Sub miniature electro-magnetic clutches and brakes are the main subjects of this 28-page booklet. Table of contents promises: Brakes; Clutch-Brakes; Duplex-Clutches; Brake-Clutches; Slip Clutch-Brakes. Catalog 957-A.
Autotronics, Inc., Florissant, Missouri.
circle number 149 on reader service card

Pulleys
Interchangeable bushings and fixed bore-pulleys with face-widths up to 148 inches, are described in catalog CV-1, newly published by Maurey. Both types described are designed for maximum service life.
Maurey Mfg. Corp., Chicago, Ill.
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Eddy current coupling
Eight-page, three-hole bulletin FAS 6, titled "Fractional Horsepower Adjusto-Speed Drive". Outlined are characteristics such as: Wide Speed Range, Low Power Losses, Accurate Speed Control, and Low Maintenance. Dynamatic Division, Eaton Mfg. Co., Kenosha, Wisc.
circle number 113 on reader service card.

Fine pitch gears
Bulletin No. 560, four-pager punched to fit covers, gives facilities and capabilities of one of the South's largest manufacturers of precision fine pitch instrument gearng. Perfecting Service Co., Charlotte, N. C.
circle number 114 on reader service card.

Belting
Data book B-220-A, a new publication describing the V-Link belting line. Complete engineering data and instructions are included in the book, describing adjustability and detachability. The Brammer Corp., New York, N. Y.
circle number 115 on reader service card.

Couplings
Bulletin 4101, an eight-page gatefold crammed with important information needed when selecting couplings. Equivalent horsepower tables, dimensions, and other facts are listed. The Falk Corp., Milwaukee, Wisc.
circle number 108 on reader service card

Clutches
Twenty-five pages of clutch applications, bound in stiff covers, hinged on the back for easy-opening. Describes special design services for every over-running clutch requirement. Formsprag Co., Warren, Mich.
circle number 109 on reader service card.

Fluid couplings
A number of folded, loose leaf pages filled with tabular data on recommended hydro - sheave drive sizes for alternating current electric motor drives. Describes a complete ready-to-install fluid transmission package. Twin Disc Clutch Co., Hydraulic Div., Rockford, Ill.
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Heavy duty sheaves
Well illustrated, 16 page booklet, describes such things as "The Results of Scientific Engineering", while at the same time listing standardized dimensions of stock, cast iron heavy - duty sheaves. The Worthington Corp., Harrison, New Jersey.
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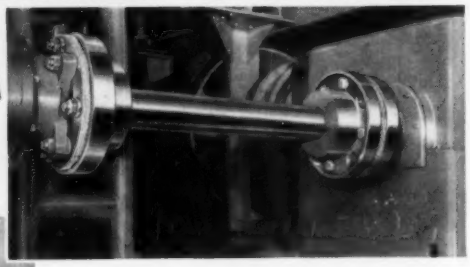

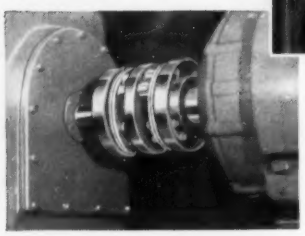

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LITERATURE *continued*

Electro-magnetic cycling

Data sheets numbered: AD-237, AD-238, AD-240, AD-243, AD-248, AD-251, give details, tables, and drawings showing cycledyne applications. Among these are: "Models with double-shaft extension;" with unitized, co-axial speed reducer on output; and others. Cycledynamics, Inc., Detroit, Mich.

circle number 112 on reader service card.

Permanent magnet d-c motor

Bulletin X-112 describes precision permanent magnet-type d-c motors designed to aircraft and missile standards. Their use and applications are outlined in detail. Globe Industries, Inc., Dayton, Ohio.

circle number 101 on reader service card.

Sheaves, belts, and pulleys

Bulletin 3101, eight-pages, covers a complete line of sheaves, variable-pitch sheaves, V-belts, flat belt pulleys, pillow blocks, flange units, take-up bearings, couplings, and timing-belt drives. T. B. Woods and Sons Co., Chambersburg, Pa.

circle number 102 on reader service card.

Gearshift drives

An eight-page technical bulletin on the gearshift drive. Describes construction and typical applications of the drives; shows original equipment photographs and gives comparative data. Dept. 489, The Lima Electric Motor Co., Lima, Ohio.

circle number 103 on reader service card.

Torque motor

New bulletin titled "Improving Performance of Flat-Armature Torque Motors" describes special techniques and drive circuits used to get superior response from this type of motor. Raymond Atchley, Inc., Los Angeles, Calif.

circle number 105 on reader service card.

Braking systems

Described are systems incorporating positive control features of: dynamic brake, uses drive motor as a generator and pro-

vides braking action; hydraulic brakes in each drive wheel give service braking; mechanical brake. used for emergency stops. Schematic drawings of brake in operation highlight the bulletin.

The Elwell - Parker Electric Co., Cleveland, Ohio.

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Variable speed transmissions

Specifications, illustrations, descriptions. Special interest is on transmission-types "A" and "D", sizes 33 to 2020; also featured are constant displacement reversible motors, types "B" and "H", sizes 1 to 60. Bulletin 10051-G,

The Oilgear Co., Milwaukee, Wisc.

circle number 123 on reader service card

Power transmission equipment

Sheaves, belts, couplings, bearings, collars, pillow blocks are listed in this eight-pager, covering complete line. Drawings, photos, tabular data give details needed by designers. Bulletin 3101.

T. B. Wood's Sons Co., Chambersburg, Pa.

circle number 124 on reader service card

Flexible couplings

Filing information. A four-page, two-color folder listing important data on flexible couplings, using resilient rubber or neoprene balls. Shown are ways of checking alignment and removing balls. Bulletin PB 11000-1-154.

Elliott Co., Crocker - Wheeler Div., Ampere, N. J.

circle number 125 on reader service card

Flexible couplings

Couplings which absorb load pulsations and shock are described in a four-page catalog-style bulletin. Features of the gear-grip include: prevention of end-thrust, universal joint action, large twist angle for taking up shock, no need for lubrication, etc. Bulletin No. C-112.

The Guardian Products Corp., Michigan City, Indiana.

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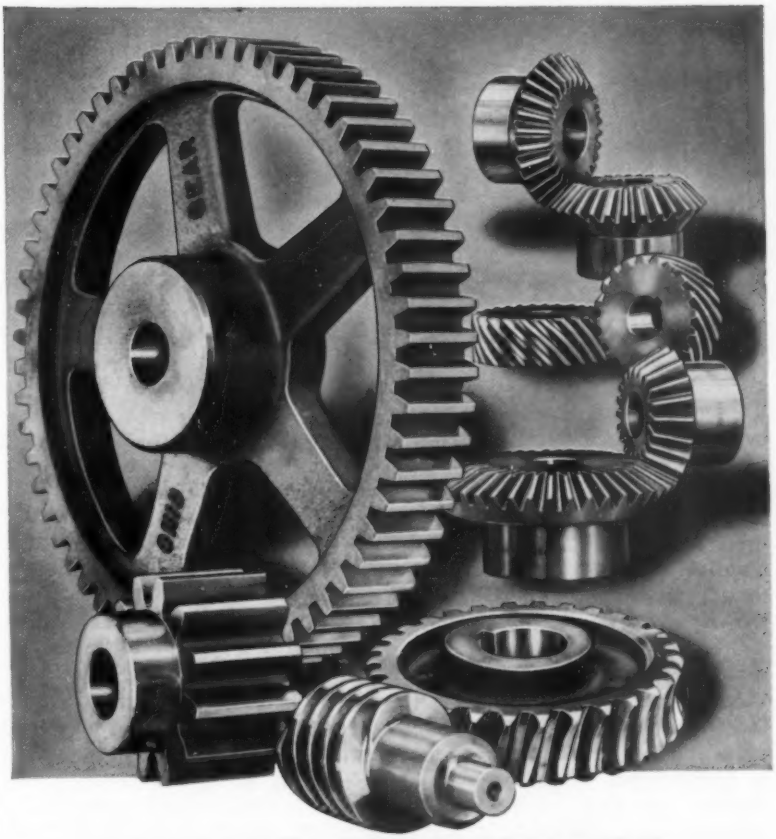
Dry fluid drives, couplings

Product photographs, engineer-



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LITERATURE *continued*

ing drawings, and installation pictures give details of dry fluid drive and coupling applications in many industries. Stock couplings and drives range from fractional hp to 1000 hp. Tables included will facilitate specification of drives and couplings. Bulletin A-640B, 24 pages.

The Dodge Manufacturing Co., Mishawaka, Indiana.

circle number 126 on reader service card

Gear reducers

Tells about such features as seven basic reducer-models covering all horsepower requirements, with each model having 30 different ratios. Other subjects covered by "High Capacity—High Quality. Swedish Style Planetary Gear Reducers" include ball tnd roller bearing construction.

Crichton Co., Johnstown, Pa.

circle number 137 on reader service card

V-belts

Greater resistance to oils, heat, abrasion, chemicals, and ozone is highlighted. High transverse rigidity of Boston V-belts is pointed out. Higher strength and stretch-resistance are described. Form No. 2012-R58.

Boston Woven Hose and Rubber Co., Boston, Mass.

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Bearings

Sixteen pages of details on ball and roller bearings in Catalog No. 28, include such specifics as: Roller Thrust Bearings; Ball Thrust Bearings; Ball Thrust Step Bearings; Journal Roller Bearings; etc.

The Gwilliam Co., Brooklyn, N. Y.

circle number 139 on reader service card

Power transmission belting

Describes Extremultus belts, features subjects: "Shock Loads", "Cone Pulleys", "Selection of Belt Thickness", "Tension and Maintenance of Extremultus", etc., in 10 pages.

Extremultus, Inc., New York. N. Y.

circle number 132 on reader service card

Angle gears

Selecting 90° power takeoffs for aircraft applications is simplified by bulletin No. AA-58, a four-pager stocked with cutaway photos, design drawings and charts. It shows major construction and design features.

Airborne Accessories Corp., Hillside, N. J.

circle number 144 on reader service card

Variable pitch sheaves

Variable pitch drives are listed in detail in this 10-page booklet, No. V-183-B. Variable speed drives with: wide range sheaves, locking ring sheaves, and two-groove locking ring sheaves are tabulated.

Browning Manufacturing Co., Maysville, Ky.

circle number 143 on reader service card

Magnetic drive

How Ampli-Speed works; How Automatic Speed Control Works. Both are revealed and explained with assistance of an excellent cut-away illustration of the magnetic drive and its components. Bulletin No. 4400-PRD-243.

Electric Machinery Mfg. Co., Minneapolis, Minn.

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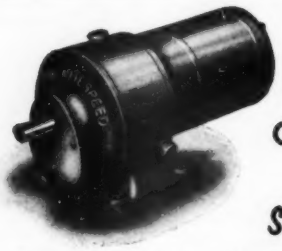

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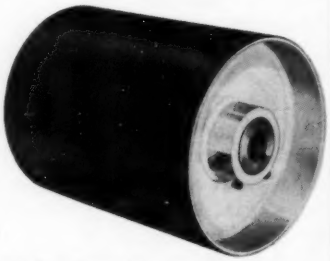

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For more information, circle No. 79 on the Reader Service Card

Motors

Industries such as: Food, Metal, Chemical, Petroleum, Plastic, Paper, Printing. and Textiles—all have special uses for electric motors. These are described in a large, double-size, unfolding brochure titled "Sterling Electric Power Drives Turn the Wheels of Industry."

Sterling Electric Motors, New York, N. Y.

circle number 145 on reader service card

Power terminal

How to save cost, space, and weight is told in this two-color, four-page bulletin, No. PPT-58, giving details of the packaged power terminal. This is because the package includes operations which were once done separately.

J. D. Christian, Engineers, San Francisco, Cal.

circle number 146 on reader service card

Motion indicator

Construction features of the Roto-Guard are shown in drawings and cutaways. These are: dust-tight switch enclosure, adjustable swich mounting, self-lubricating ball bearing construction, and others.

The Bin-Dicator Co., Detroit, Michigan.

circle number 147 on reader service card

Speed reducer

Lists several of many applications for the Shaftex speed reducer. Included are: pumps, machine tools, runout rolls, rotary screens, mixers, tumblers, agitators, conveyor systems. hoists, and others. Bulletin 20 E 8903.

Allis Chalmers Manufacturing Co., Cincinnati, Ohio.

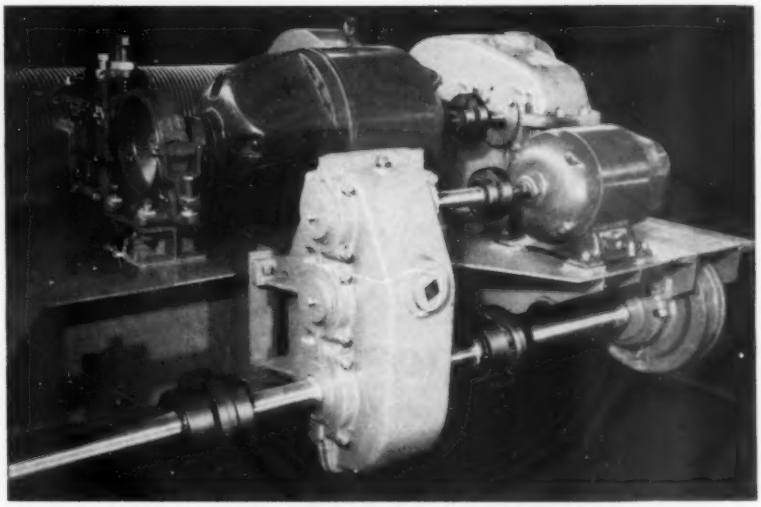
circle number 141 on reader service card

Controls

An 8-page descriptive bulletin, No. 958, illustrates industrial controls. Both individual products and assembled controllers are illustrated and described. Also featured are several speed-responsive switches.

The Euclid Electric and Manufacturing Co., Madison, Ohio.

circle number 142 on reader service card

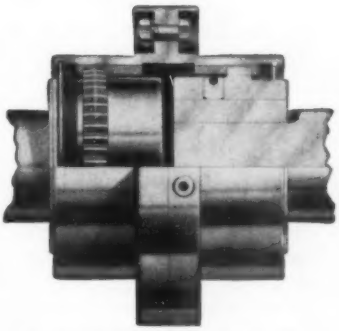


AJAX 3-D Dihedral Couplings with 3-degree capacity relieve strains between motors, gear boxes, and traction wheels.

FOUR AJAX 3D DIHEDRAL COUPLINGS ELIMINATE MISALIGNMENT HAZARDS ON EUCLID CRANE AND HOIST TROLLEY

Crane service is hard service, Heavy loads, constant starting, reversing, and braking set up unavoidable weaving and twisting stresses on all components.

The photograph shows how Euclid Crane & Hoist Co., Cleveland, Ohio safeguards performance by protecting bearings, gears, armatures, frames, and housings against misalignment strains by using AJAX 3-D Dihedral Couplings. Located between all driving and driven units of trolley and hoist, the four AJAX 3-D Dihedral Couplings insure against costly interruptions to crane service and difficult expensive repairs due to inaccessible overhead installations.



Cutaway section of AJAX 3-D Dihedral Coupling with 3-Degree angular and/or offset capacity. Other standard Dihedral Couplings up to 7 degree capacity.

Also AJAX Bronze bushed Rubber cushioned Couplings.

The patented Dihedral tooth design of AJAX Dihedral Couplings reduces backlash to the vanishing point, an important factor in the smooth operation and precise control essential to load handling.

The nationwide organization of AJAX Coupling specialists offers broad and fully qualified experience as an aid in improving performance and reducing costs.

AJAX FLEXIBLE COUPLING CO. INC.
WESTFIELD, N. Y.
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For more information, circle No. 27 on the Reader Service Card

ABSTRACTS

of current technical papers

Total composite error and position error

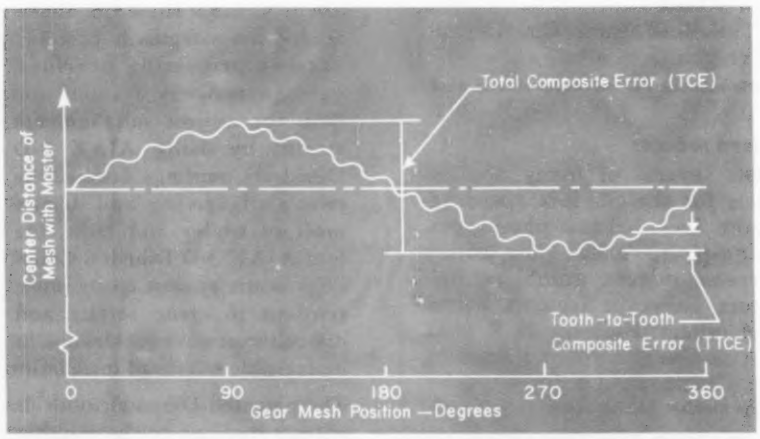
By **GEORGE W. MICHALEC,**
Section Head, Engineering Division, General Precision Laboratory, Inc.

THE DESIGN AND FABRICATION of precision gears has led to the development of total composite error, TCE, and angular position error. TCE is widely understood. Angular position error is a more recent idea and is still limited to high precision work.

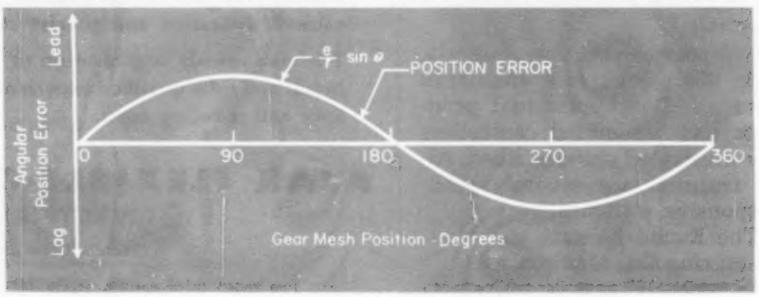
Position error is sometimes called index error, but committees of the American Gear Manufacturers Association (AGMA) prefer the term "position error."

Total Composite Error

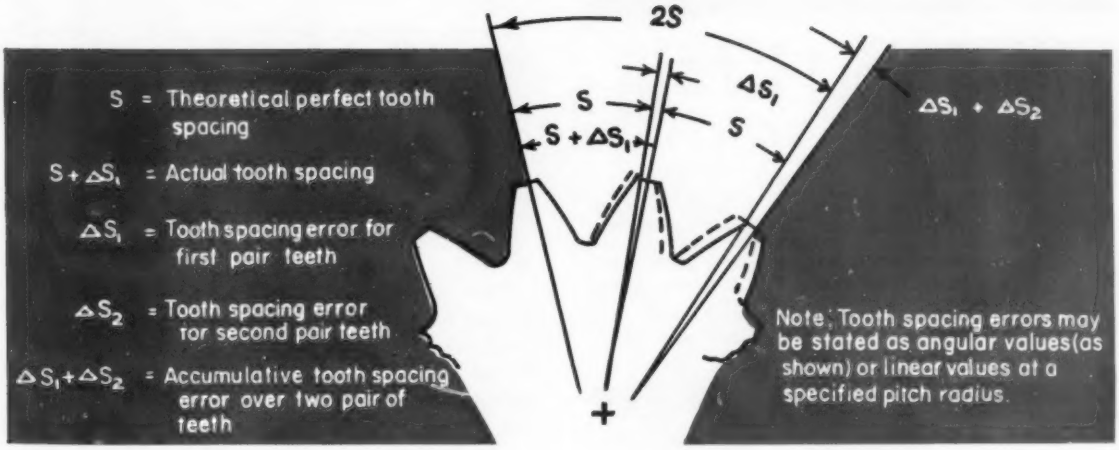
It includes tooth-to-tooth variation due to profile error, lead error, pitch error, and tooth spacing error due to non-concentricity of the center of rotation with the pitch circle, and secondary effects due to lateral runout or side wobble.



TOTAL COMPOSITE error is the maximum range of center distance varation of a gear when rolled with a master gear on a variable center distance fixture.



POSITION ERROR is caused by inaccuracy of tooth spacing in either circumferential or radial direction.


2S
S = Theoretical perfect tooth spacing
S + ΔS1 = Actual tooth spacing
ΔS1 = Tooth spacing error for first pair teeth
ΔS2 = Tooth spacing error for second pair teeth
ΔS1+ΔS2 = Accumulative tooth spacing error over two pair of teeth
S
S + ΔS1
ΔS1
S
ΔS1 + ΔS2
Note: Tooth spacing errors may be stated as angular values (as shown) or linear values at a specified pitch radius.

Ability of gears to transmit motion smoothly and with constant velocity is directly related to TCE. Consider the effects of the major components of TCE: runout and tooth-to-tooth composite error. Runout in a gear causes a sinusoidal positional error **E**

$$E = \pm \frac{e}{r} \sin \theta$$

where e = eccentricity or ½ runout
r = radius of gear
θ = angle of mesh position around gear

The other major component of TCE is tooth-to-tooth composite error. It includes spacing error, profile error, lead error, and tooth thickness error.

Tooth-to-tooth composite error cannot be greater than the TCE and is usually less. Therefore, position error due to tooth-to-tooth composite error will never exceed the positional error if cumulative effects are disregarded.

Position Error

Position error is caused by inaccurate tooth spacing. Theoretically, teeth are to be concentric with the axis of rotation and equally spaced along the pitch circle. In practice, there are imperfections. Inaccuracy of the profiles is due to an integration of all tooth-to-tooth errors and eccentricity of rotation. This results in profile displacement from the ideal position.

Profile position error is related to the total composite error, but the relation is not always simple and direct. It does directly affect angular position accuracy of mated gears. The angular positions would be relative to the axis of the gear, which may be either a bore or a journal diameter.

Position error inspection is random. Usually the angular position of the tooth profile is checked by probing the tooth contour or inserting a probe between the teeth. Because angular position errors are small, the angle of position measurement is often checked with optical instruments. It is a slow and expensive operation.

Significance of Position Error

For 10 years the gear industry has been using TCE to gage gear quality, particularly the angular positional accuracy of transmission. This has been a great achievement, compared to the days before World War II when gears were merely specified by pitch diameter and tolerance. A gear can meet TCE specifications and yet have severe position error. The converse is also true.

Improper Number of Teeth

A limitation of TCE inspection is established by considering a gear made with one less, or one more tooth than specified. If the gear were otherwise perfect, the TCE would be within specification yet the teeth could have huge position error. Consider a 47-pitch gear, 96 teeth, having only 95 teeth. Each tooth would be in position error by about 0.0005". This is a relatively small error, but since it is accumulative, it amounts to one tooth and space 0.0654" around the gear. In making TCE inspection on a variable center distance master rolling inspection device, the values of center distance for a complete rolling of the gear could be perfect and constant since the tooth spacing is constant, although in error. For proper cutter setting, the center distance error, when rolled with a master, would only be in the order of 0.001". This would not be taken as a clue showing the gear had the wrong number of teeth. Alternately, there would not be any center distance error if the gear cutter is set to proper depth by use of the center distance inspection data. The easy way to detect the flaw is to count the teeth.

Relations

Although TCE and position error are different, they are related. Both indicate the degree

Continued on page 86

UNBALANCED THRUST LOADS

Values used to compute factors in table 3

Total no. of pistons N	Even number of pistons loaded: No. of loaded pistons N_1	$\frac{a}{R}$	λ	F_1	Odd number of pistons loaded: No. of loaded pistons N_1	$\frac{a}{R}$	λ	F_1
5	2	.8090	84.10	3.340	3	.5393	154.20	2.089
6	2	.8660	68.96	4.018	3	.6667	118.50	2.487
7	4	.5617	146.57	2.146	3	.7490	98.74	2.895
8	4	.6533	121.80	2.435	3	.8047	85.21	3.301
9	4	.7199	105.70	2.731	5	.5759	142.29	2.184
10	4	.7694	93.87	3.026	5	.6472	123.27	2.413

Table 1

When a/r is 0.5 or less $F = 1 + 2\,a/r$

a/r	.50	.55	.60	.65	.70	.75	.80	.85	.90
λ	180.0	150.4	135.5	122.6	110.5	98.6	86.4	73.5	59.0
F_1	2.00	2.115	2.254	2.423	2.631	2.899	3.260	3.785	4.661

Table 2

No. of pistons	5	6	7	8	9	10
F	5.25	6.31	6.82	7.78	8.58	9.51

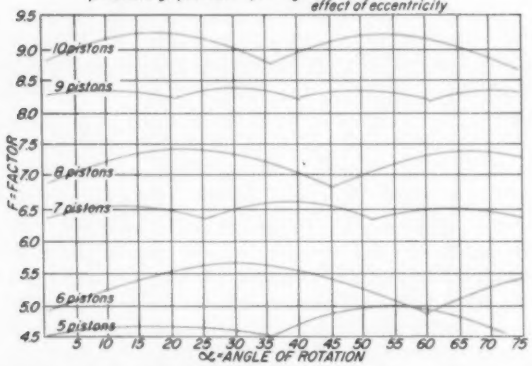
Table 3

USE these tables with the formulas.

Effective axial load factors in axial piston pumps plotted against angle of rotation

$P = F p d^2$

P = Total effective load – lbs; d = Diameter of pistons – in
F = Factor = $\frac{\pi}{4} n_1 F_s$; n_1 = No. of pistons loaded
p = Discharge pressure – psi; F_s = Correction factor for effect of eccentricity



CURVES OF FACTORS for all angular positions of shaft through 75° rotation.

Dimension a must be the same for both thrust bearing and pistons. Value of a for any desired pump can be found by using value of a/R from table 1 and computing a/r, obtaining the correct factor F_1 from table 2. If a/r is less than 0.5, then F_1 can be computed.

$$F_1 = 1 + 2\frac{a}{r} \qquad (14)$$

Derivation of F_1 when a/r is less than 0.5 can also be obtained.

In most pumps the pitch diameter of the pistons and thrust bearing are the same, or nearly so. When this is the case, values for F_1 for each condition can be computed in table 1. Calculations are further simplified

$$P_e = F_1 \frac{\pi d^2}{4} \frac{p}{\cos \delta} N_1 \qquad (15)$$

$$F = \frac{\pi}{4} N_1 F_1 \qquad (16)$$

Equation (15) becomes

$$P_e = F \frac{p}{\cos \delta} d^2 \qquad (17)$$

The use of corresponding pairs of values of N_1 and F_1 from table 1 will determine the worst condition for any probable number of pistons. These conditions are indicated by asterisks. Using only the worst conditions will produce the values in Table 3. Equation (17) provides a simple method of computing maximum effective load when the two pitch diameters are approximately equal.

The tables give the proper factor for the worst conditions only. The curves show factors for all angular positions of shaft through 75° of rotation. It illustrates that effective bearing load is more uniform for an odd number of pistons than for an even number.

Sometimes the inlet pressure is not zero as stated in assumption 1. Assume a pump with seven 0.750 in. diameter pistons on a 2.125 in. pitcch diameter. Wedge angle is 13°. Inlet pressure is 400 psi and discharge pressure is 950 psi.

Whenever inlet pressure is more than 1% of outlet pressure, critical loading on the bearing occurs when N/2 or (N+1)/2 pistons are subjected to outlet pressure. Since inlet pressure must be considered,

$$P = \frac{\pi d^2}{4} \frac{p}{\cos \delta} \frac{N+1}{2} + \frac{\pi d^2}{4} \frac{p_1}{\cos \delta} \frac{N-1}{2} \qquad (3a)$$

$$P\,a = \frac{\pi d^2}{4} \frac{p}{\cos \delta} \times 2 \left(\cos \frac{\theta}{2} + \cos \frac{3\theta}{2} \right) R - \frac{\pi d^2}{4} \frac{p_1}{\cos} \left(1 + 2 \cos \theta \right) R \qquad (5a)$$

Simplifying (3a) and (5a) and substituting the values from this example gives

$$P = \frac{\pi (.750)^2}{4} \frac{1}{\cos 13^\circ} \left(950 \times 4 + 400 \times 3 \right)$$

$$= 2270 \text{ lbs.}$$

$$P\,a = 2270a = \frac{\pi (.750)^2}{4} \frac{2.125}{2 \cos 13^\circ} \Big(2{,}950 \, (\cos 25.71^\circ + \cos 77.14^\circ) - 400 \, (1 + 2 \cos 51.43^\circ) \Big) = 595$$

Therefore a = 0.263 in.

Due to appreciable inlet pressure, the effec-

(*Continued on page 86*)

BOOKS

Gear Ratio Tables for 4- 6- and 8-Gear Combinations

By Earl Buckingham, Brighton, England; Published by the Industrial Press, 93 Worth St., New York, N.Y.; Library of Congress catalog card No. 58-13792. Price: $5.00.

Contains tables of consecutive, factorable numbers from 1000 to over 40,000, with values of their ratios and reciprocals of ratios.

The book shows by specific examples how these tables may be used to obtain suitable sets of change-gears. Divided into four parts, the first section gives examples worked out, showing how calculations are done.

Parts two and three are made up of tables of consecutive factorable numbers with ratios and reciprocals of ratios. Brocat's tables of ratios make up part four.

Introduction to Electrical Applied Physics

By N. F. Astbury M.A., Sc.D., M.I.E.E., F. Inst. P, Professor of Physics, University of Khartoum; Published by the Philosophical Library, 15 E. 40 St., New York 16, N. Y.; Price $4.75.

Astbury's essay is an excellent attempt to bring together subject and specialist topics.

Treatment of chosen topics has been circumscribed somewhat by lack of space. But, in essence, the book is not only a matter of new work accomplished, but also reflects a revived interest in applications of earlier work.

Chapter and section headings include titles like: Electromechanical Coupling; Electrical Units and the General Equations; The Magnetic Field; Electromechanical Systems with Feed-Back; Some Mechanical Considerations.

Industrial Lubrication Practice

By Paul D. Hobson, P.E., A.M.I. Mec. E., A.M.I. Mar. E.; Research Associate, Pennsylvania State University; Published by Industrial Press, New York, N. Y., Price $8.00.

Guide to a sound lubricating-practice and its application as a cost-saver in machine operation and maintenance.

This practical working manual covers motors, engines, and generators. Titles listed in the table of contents include: Running in New Bearings; Plain Bearings; Anti-Friction Bearings; Methods of Lubricant Supply; Storage Preservation of Machinery; Chains, Ropes, Belts, and Couplings; and Gears.

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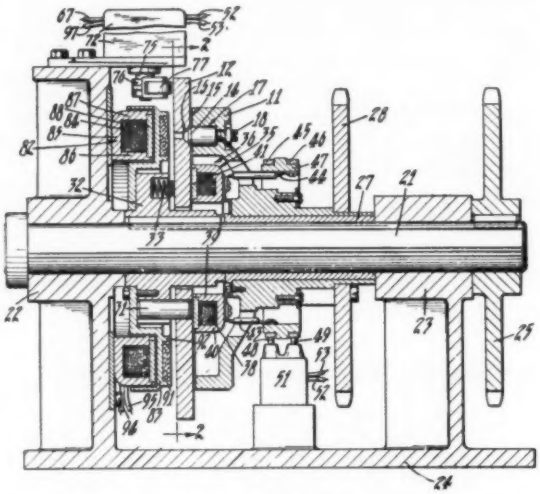
PATENTS

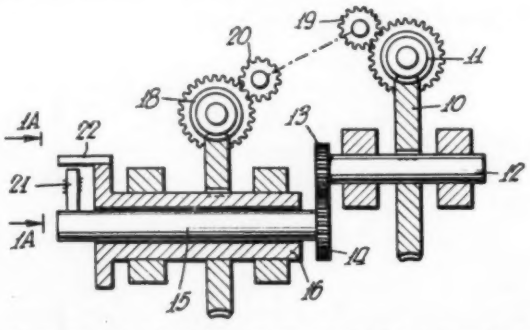
Release Clutch

U. S. Patent 2,856, 046; Karl Bofinger, Cincinnati, Ohio, assignor to American Can Co., New York, N. Y.



This unit has a driving clutch member and a driven slip clutch plate. Slippage of the plate against the driving member performs the disengaging function. A stationary magnet is mounted adjacent to the path of travel of the plate. An attached member moves with the plate and presses between the plate and magnet to receive the magnetic forces and to attract and hold the plate disengaged from the opposing member.

Gear Accuracy Tester



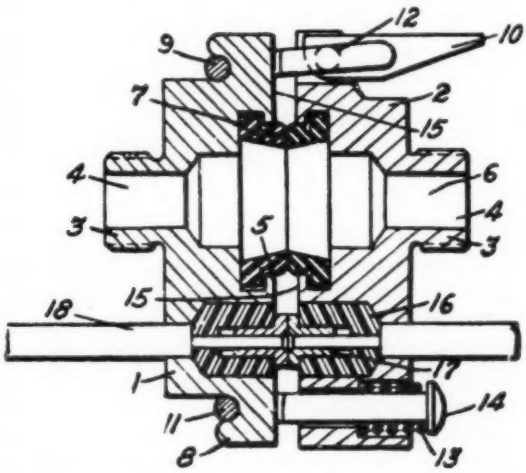
U. S. Patent 2,855, 691; F. W. Cunningha, Stamford, Conn.

This tester tests accuracy of gears. It has two

shafts with meshing gears. One gear is the test gear and the other a high accuracy gear. A third shaft is coaxial with the second with a worm gear powering the first. The third shaft is driven by a separate worm gear drive. The first and third shafts operate in reverse ratio of the nominal pitch diameters of the first and second gears. The test gear is mounted on either the first or third shaft and, if accurate will rotate satisfactorily with the coaxial second and third shafts.

Coupling

U. S. Patent 2,857,576; Ernst U. Mannheim, Germany, assignor to Knorr-Bremse, Munich, Germany.
This coupling has two opposing flanges with fluid ducts in each leading to the engaging surfaces. There are rubber rings projecting from the surfaces of each flange. The rubber rings are pressed tightly against each other to form a fluid tight seal. There are elec-



trical contacts beside each ring forming electrical continuity for current. A spring-loaded retainer and toggle mechanism keeps the two parts in coupled position. The retainer and toggle are mounted on one flange and a hook is on the other. The hook engages the toggle. A second hook on the second flange is engaged by the retainer.

Shaft Seal

U. S. Pattent 2,857,182; William R. Bain, Frederick W. Hoeltje, Wylie L. Johnson and John B. Wheatley, Indianapolis, Ind., assignors to General Motors Corp., Detroit, Mich.
This patent is based on a fluid seal of the sliding-contact type which has a support member and a cage that holds a shaft than can rotate but yet has freedom for axial movement. A seal follower is mounted on the shaft adjacent to the bearing. A sleeve encircling the shaft is adjacent to the follower and recipro-

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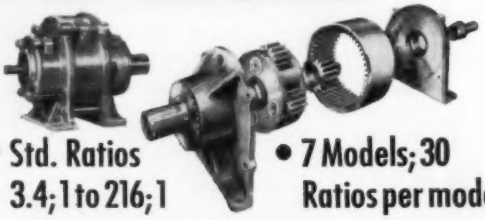

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COMPUTING MOTORS *continued from page 33*

energy. When you choose a motor, take care to insure that Δw is within the operating speed range.

The method for calculating time required by the motor to accelerate the load and flywheel back to original speed is more difficult. A method for making this calculation is: Let T_{m1} equal the motor-torque obtained from the speed-torque curve at the beginning of the acceleration period, and T_{L1} equal the load torque at the same instant. Let speed of the motor in rpm at the beginning of the acceleration period be N_1. The available accelerating torque at the beginning of the accelerating period is

$$T_1 = T_{m1} - T_{L1} \text{ Lb-ft}$$

Let T_{m2} equal the available motor torque at the end of the accelerating period (load and flywheel at normal speed) and T_{L2} equal the load torque at the same instant. The speed of the motor in rpm at the end of the accelerating period is N_2. The available accelerating torque at the end of the accelerating period is

$$T_2 = T_{m2} - T_{L2} \text{ Lb-ft}$$

the latter is a point of stable operation, thus motor torque equals load torque, T_2 becomes zero.

$$\text{Let } T = \frac{T_1 - T_2}{2} \text{ Lb-ft}$$

$$A = T_1/T_2$$

$$T_e = KT$$

Where K is a constant which varies as a complex function of a (see table).

Time in seconds required for the motor to accelerate load and flywheel can be calculated

$$t = \frac{J\,(N_1 - N_2)}{9.55\,T_e} \text{ sec}$$

Acceleration time must be shorter than the length of the duty cycle, or if the duty cycle has more than one peak it must be shorter than the period between peaks.

PATENTS *continued from page 81*

cates in the cage. The seal annulus engages with the follower. A ring seals between the support member and the sleeve member. The seal ring has one of its peripheral surfaces engaged with a cylindrical surface and is relieved over a portion of its peripheral axis. The surfaces are exposed to fluid pressure which forms an abutment on the shaft when the annulus is in contact with the follower. This moves the sleeve toward the follower when the shaft moves axially. The shaft carries lubricating oil under pressure into the follower. Ports in the follower discharge oil against the face of the annulus and other ports discharge oil against the shaft, forming a seal. Additional ports discharge oil into the cage which has a drain.

from conception, when pegs and spokes were used to transmit motion, through the cycloidal tooth form and finally the present day involute tooth form systems.

The Tooth Form Committee, chairmaned by G. P. Maurer, Falk Corporation, reviewed a revised copy of standard AGMA 201.02, Tooth Proportions for Coarse Pitch Involute Spur Gears. The revised draft incorporates the proposal covering standardized tip relief of the basic rack tooth form.

The Nomenclature Committee, chairmaned by Granger Davenport, Gould and Eberhardt Co., is revising standard AGMA 110.02, Nomenclature of Gear Tooth Wear and Failure. The committee is also planning to revise and combine standards AGMA 111.03, Letter Symbols for Gear Engineering, AGMA 112.03, Gear Nomenclature, and AGMA 116.01, Abbreviations for Gearing.

The Aircraft Engine and Power Take-off Gearing Committee, chairmaned by John C. Leming, Gear Grinding Machine Co., has completed work on proposed tentative standard AGMA 431.01, Design Procedure for Aircraft Engine and Power Take-off Bevel Gears. The committee has started work on a new scoring formula.

The Wormgearing Committee, chairmaned by J. E. Gutzwiller, DeLaval Steam Turbine Co., is working on a revision to standard AGMA 341.01, Design of Coarse-Pitch Cylindrical Wormgearing.

The Aircraft and Missile Control Gear Committee, chairmaned by C. A. Underwood, Raytheon Mfg. Co., has just completed and approved proposed tentative standard, Control Gear Accuracy Specifications. The scope of this standard includes: a) definition of gear errors as applied to control gears; b) methods of measuring gear tooth errors and gear blanks; c) procedure for the com-

continued on next page

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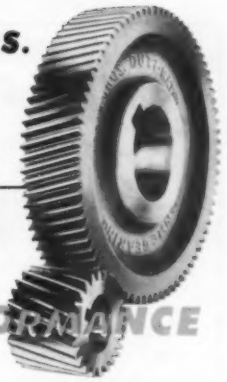
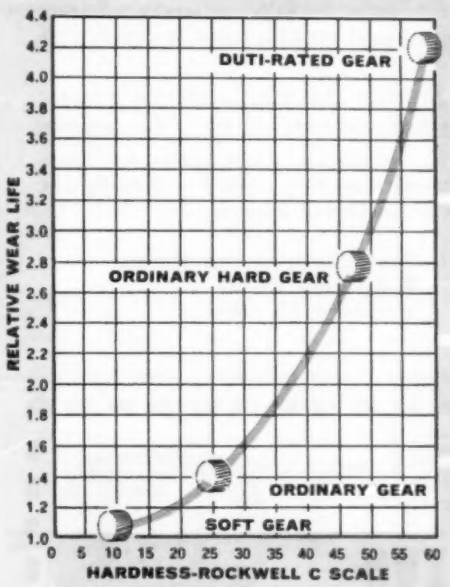


GEAR PERFORMANCE COMPARISON CHART



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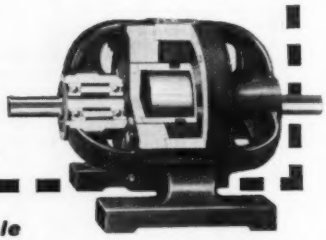

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MEETINGS

JANUARY 26-27

American Society of Lubricating Engineers, Gear Symposium, Morrison Hotel, Chicago, Ill. Contact: Calvert Willey, Executive Secretary, 84 East Randolph St., Chicago, Ill.

JANUARY 28-29

Association of Roller and Silent Chain Manufacturers, annual meeting, Drake Hotel, Chicago, Ill. Contact: A. L. Taylor, Executive Secretary, Upton Station, PO box 55247, Indianapolis, Ind.

AGMA MEETING *continued*

pensation of measuring errors; d) accuracy levels for control gears; e) effective size.

The Bevel Gearing Committee, chairmaned by Wells Coleman, Gleason Works, is working on combining standards AGMA 206.03, Fine-Pitch Straight Bevel Gears, and AGMA 208.01, 20-Degree Straight Bevel Gear System, into one standard. A task Committee was appointed to act with the Gear Rating Committee to review bevel gear strength and durability standards.

The Metallurgy and Materials Committee, chairmaned by John Erler, Farrel-Birmingham Co., Inc., is working on a revision to standard AGMA 241.01, Gear Materials—Steel. The committee approved the release of tentative standard AGMA 244.01, Gear Materials—Nodular Iron. This standard will be available shortly from AGMA headquarters.

The Measuring Methods and Practices Committee, chairmaned by F. L. Heine, Milwaukee Gear Company, has appointed a Task Committee Chairman for each of the following projects:

a) Rolling Method of Inspection
b) Individual Error Inspection
c) Calibration of Master Gears

d) Indicators
e) Inspection Equipment
f) Measurement over wires Inspection
g) High Production Coarse Pitch Gear Inspection

The committee plans to cover all areas of gear inspection. This data most likely will be published in the form of Information Sheets.

The Fine Pitch Gearing Committee, chairmaned by P. M. Dean, Jr., General Electric Co., has completed proposed tentative Information Sheet AGMA 114.01, Formats for Gear Specification Data. This Information Sheet is intended to form a common language bridge between design, manufacturing inspection, buyer and seller. The committee is also working on a backlash standard, Lost Motion Compensation.

The Cutting Tools Committee, chairmaned by Ben F. Bregi, National Broach & Machine Co., is working on a standard for oversize wormgear hobs. The committee is also preparing to write a standard on gear shaper cutters. They will begin soon.

The Splines and Keyways Committee, chairmaned by Harry Pelphrey, Michigan Tool Co., is actively preparing a revision to standard AGMA 261.01, Keyways for Holes in Gears. The revision will constitute a comprehensive treatment of standard keys and keyways from the crudest to the finest, and woodruff keys. Research for this project has been carried out on company, national and international levels.

The Lubrication Committee, chairmaned by Mr. J. H. Allen, Farrel-Birmingham Co., Inc., has just completed proposed tentative standard AGMA 252.01, Mild Extreme Pressure Lubricants for Industrial Enclosed Gearing. This standard will be available shortly from AGMA Headquarters. The committee held a joint symposium with the National Lubricating Grease Institute on Lubrication of Industrial Gearing. Mr. J. H. Allen, and Mr. J. Hitchcock presented papers for AGMA.

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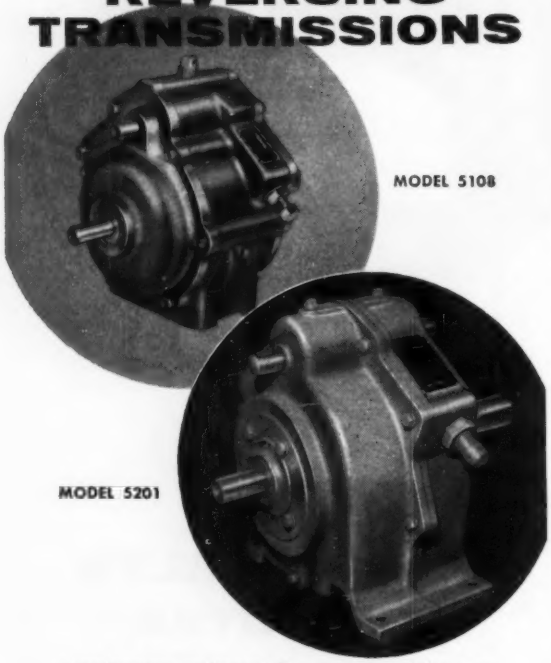

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GEAR ERROR *continued*

of departure of a gear from perfect positional accuracy. For example, if a gear has a specified maximum position error, what will be the equivalent maximum TCE, and vice versa. A helpful way to illustrate equivalents is to consider special cases.

A gear which has a TCE predominantly due to eccentricity error will have a sinusoidal TCE trace. The position error will be greatest when the true geometric center of the gear is 90°, either side of the line of centers. The maximum position error will be the total spread from peak to peak, approximately

$$E_T = \frac{TCE\ (\text{radians})}{R} \qquad (2)$$

Consider a maximum position error of 165 sec of arc and a pitch radius of 1.250". The equivalent TCE, assuming all eccentricity error, will be

$$TCE = R\,E_T = \frac{1.250 \times 165}{60 \times 60 \times 57.3} = 0.001 \text{ in}$$

A TCE which is entirely composed of tooth-to-tooth error will cause position error because of tooth size and spacing variations. The equivalent TCE can be determined by calculating how much the center distance is altered in a variable center distance check for profile errors in adjacent teeth.

$$TCE = \Delta CD = \frac{\Delta S}{2 \tan \theta} \qquad (3)$$

where

ΔCD=change in center distance

ΔS=Tooth-to-tooth linear position error

θ=pressure angle

Total angular position error is

$$E_T = \frac{\Delta S}{R} = \frac{TCE \times 2 \tan \theta}{R} \text{ radians} \qquad (4)$$

In actual practice, ΔS may be due to profile deviations, pitch error or combinations. As an example, for a position error of 120 seconds, R=1.250, θ=20°, using formula 4, the equivalent TCE = 0.001 inches.

The more general case is for a gear to have TCE composed of both runout and tooth-to-tooth error. For commercial quality gears the runout component usually is predominant, but for high precision gears the tooth-to-tooth error becomes more significant and may often be the greater component error. Should it be desired to express a given position error in terms of a TCE which is composed of an assumed ratio of runout and tooth-to-tooth error, formulas 2 and 4 can be combined to express the TCE value

$$E_T = \frac{a\,(TCE)}{R} + \frac{b\,(TCE \times 2 \tan \theta) \text{ radians}}{R} \qquad (5)$$

where

a=fraction of TCE which is runout.
b=Fraction of TCE which is tooth-to-tooth error.
$a + b = 1$

The two terms in equation (5) are similar, particularly for 20° pressure angle or greater. Therefore, since position accuracy involves many approximations, gear engineers often disregard the breakdown of equation (5) apply equation (2) to the entire TCE, and still get good results.

How To Specify

Because a TCE check can't reveal all position errors, design engineers should control position-error. This is done on gear drawings by calling for maximum allowable error between any two points on the same or different profiles. The specification is usually angular, in seconds or minutes of arc.

When specifying maximum position error it is still possible to also specify TCE and TTCE without being redundant. As long as this specification is consistent with the specified position error there will be no incompatibility.

At present there are no standards for specifying gear position error. However, a sub-committee of AGMA Aircraft Gear Committee is preparing a specification which considers position errors for control gears.

In applying the facts of this article, remember they are for no-load conditions. Load deflections must be separately considered. Beside gear teeth, you should consider such things as hubs, bearings, shafting, and housing.

Causes

Except for eccentricity errors due to runouts of the gear blank, generating tool, work arbor and tool arbor, the error sources are associated with vibrations, tool chatter, material deflection, tool imperfections, tool wear, and:

(1) Eccentric mounting of gear blank
(2) Eccentric mounting of generating tool
(3) Runout in gear blank mounting arbor
(4) Runout in tool arbor
(5) Lead error in hobs
(6) Tooth spacing errors in shaper cutters
(7) Position errors in gear generator's driving gear train
(8) Profile errors in tool
(9) Variables due to vibrations and chatter
(10) Deflections due to workload in generating
(11) Non-homogeneous gear blank material
(12) Effect of approximating involute profile with generated straight cuts
(13) Temperature effects due to profiles being cut at elevated temperature

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This article is abstracted from a paper presented at the semi-annual meeting of the American Gear Manufacturers' Association, Edgewater Beach Hotel, Chicago, October 27, 1958.

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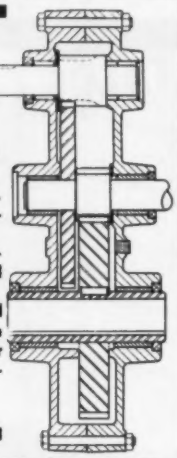

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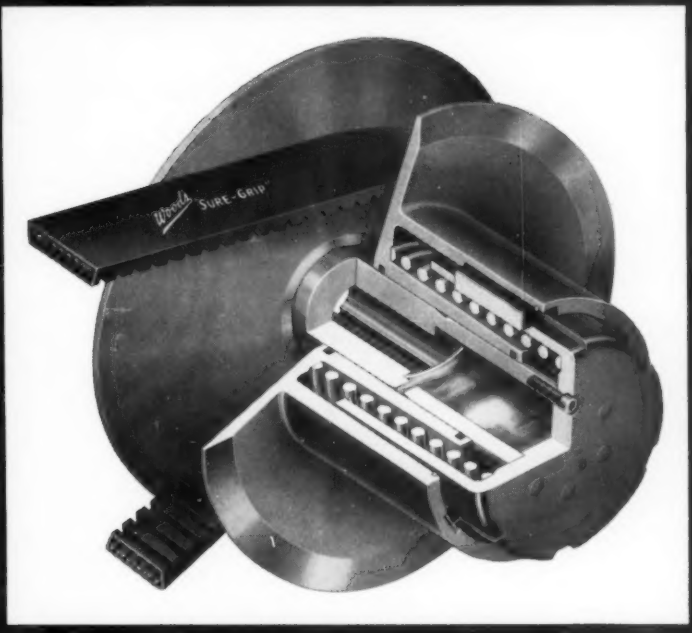

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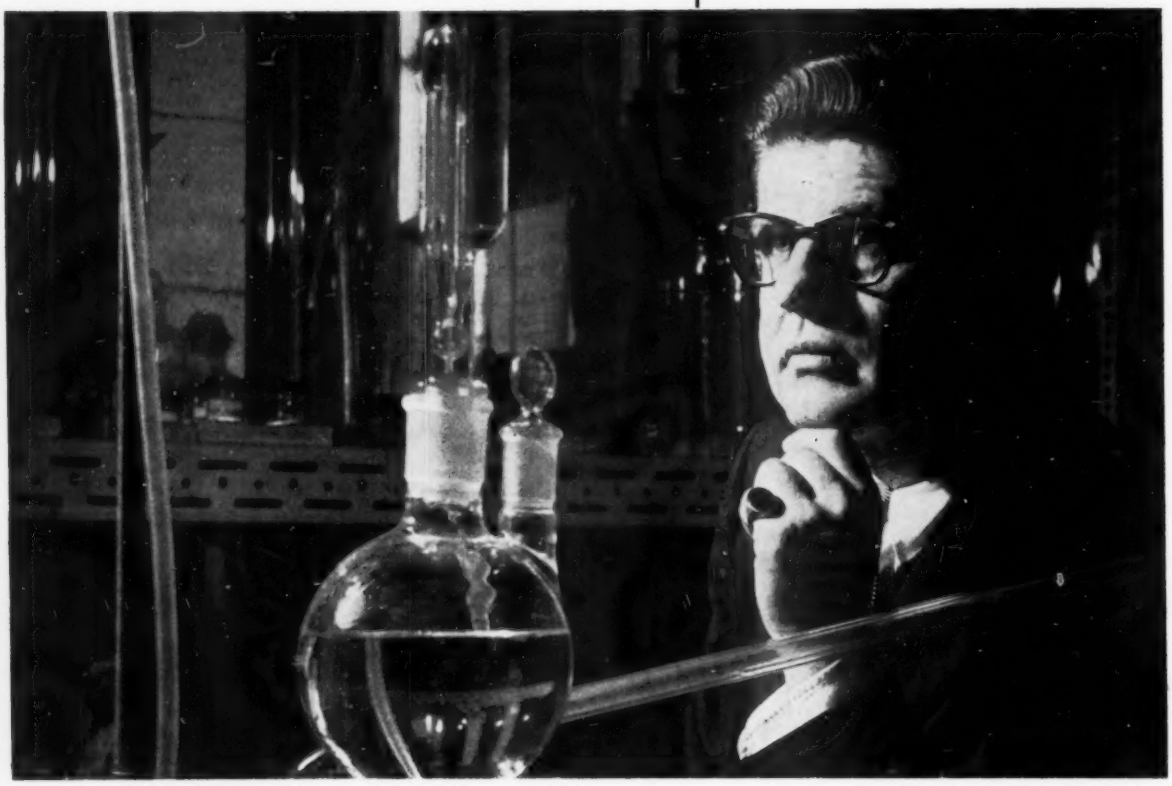

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